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RESEARCH MEMORANDUM

PRELIMINARY EVALUATION OF THE PERFORMANCE OF A UNIFLOW TWO-STROKE-CYCLE SPARK-IGNITION ENGINE COMBINED WITH A BLOWDOWN TURBINE AND A STEADY-FLOW TURBINE

By Bernard I. Sather and Hampton H. Foster

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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PRELIMINARY EVALUATION OF THE PERFORMANCE OF A UNIFLOW TWO-STROKE-CYCLE SPARK-IGNITION ENGINE COMBINED WITH A BLOWDOWN TURBINE AND A STEADY-FLOW TURBINE

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SUMMARY

Calculations based on a theoretical analysis were made for a composite engine consisting of a uniflow two-stroke-cycle spark-ignition engine, a compressor, a blowdown turbine, and a steady-flow turbine. Values of net brake specific horsepower and fuel consumption are presented for a wide range of compression ratios and ratios of exhaust pressure to inlet pressure, a fuel-air ratio of 0.067, an engine speed of 2000 rpm, a maximum cylinder pressure of 1200 pounds per square inch, an inlet-manifold temperature of 200° F, and an altitude of 30,000 feet. The total air flow and power of the component engine were based on test data from both two- and four-stroke-cycle engines.

Operation of the composite engine was considered for the following cases:

1. Maximum available temperature of gas mixture to steady-flow turbine unlimited

2. Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F

3. Maximum available temperature of gas mixture to steady-flow turbine unlimited; blowdown turbine omitted

4. Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F; blowdown turbine omitted

The results indicate that the highest net specific powers are obtained at the lowest component-engine compression ratios; the lowest net specific fuel consumptions are obtained at the highest compression ratios. With the component engine considered, the maximum specific

output and the minimum specific fuel consumption occur at a ratio of exhaust to inlet-manifold pressure of approximately 0.9. Exhaust-gas-temperature limitation and removal of the blowdown turbine adversely affect power output and efficiency. Where exhaust temperature is not the limiting factor, excess air above the minimum required for adequate scavenging lowers the efficiency but improves the power output.

INTRODUCTION

When the conventional reciprocating aircraft engine is examined for possible improvements in power and fuel consumption, the greatest gain seems to be offered by recovery of the waste energy in the exhaust gases and by an increase in the air-handling capacity of the engine. The exhaust-gas energy may be recovered by exhaust-jet propulsion or by exhaust-gas turbines.

Certain combinations of a reciprocating engine with exhaust-gas turbines and engine- or turbine-driven compressors, called composite engines, have been considered by various investigators. References 1, 2, and 3 discuss various types of composite engine wherein the reciprocating-engine component is a four-stroke-cycle spark-ignition engine. References 4 to 7 consider composite engines with a four-stroke-cycle compression-ignition engine as the reciprocating component.

A preliminary evaluation of a composite engine consisting of a uniflow two-stroke-cycle spark-ignition engine geared with a blowdown turbine and a steady-flow turbine in series with a suitable air compressor or compressors was made at the NACA Cleveland laboratory.

A two-stroke-cycle component engine was considered in this evaluation because the two-stroke-cycle engine handles more air for a given cylinder size than the four-stroke-cycle engine. The increased air-handling capacity of the two-stroke-cycle engine is due to two characteristics: (1) A power stroke occurs every crankshaft revolution; and (2) there is a large valve overlap during the scavenging period. The large quantity of excess air used in scavenging is useful for engine cooling and for reducing the temperature of the gases entering the steady-flow turbine as well as for increasing the mass flow through the system.

The two types of turbine are used in order that both the kinetic energy of blowdown and the steady-flow energy of the exhaust gases may be recovered. Experimental results for a blowdown turbine used to recover kinetic energy from the exhaust gases of an aircraft engine are reported in reference 8.

The effects of compression ratio and ratio of exhaust pressure to inlet pressure on net brake specific horsepower and fuel consumption of the composite engine were computed and plotted to find the most desirable operating conditions. The results, in the form of graphs, are shown mainly for one altitude, one engine speed, one maximum cylinder pressure, one inlet-manifold air temperature, and one fuel-air ratio. In addition, the effects of changes in inlet-manifold air temperature, scavenging ratio, and fuel-air ratio on the net performance are shown. Power output of the composite engine could be increased considerably by increasing the engine speed, by increasing the allowable maximum cylinder pressure, and by burning extra fuel in the duct leading to the steady-flow turbine. No calculations are presented, however, to show the effects of these changes. Two temperatures of the gas mixture to the steady-flow turbine are considered: a limited maximum and the maximum available. Results are shown for operation with and without the blowdown turbine. All turbines and compressors are assumed to operate at their peak efficiency under all conditions of operation.

Because this analysis is primarily a thermodynamic-cycle analysis, sizes and weights of the various components and methods of control were not considered.

ASSUMPTIONS AND METHODS OF ANALYSIS

The analysis of the composite engine involves the significant characteristics of two-stroke-cycle engines, the operation of the turbines, and assumptions necessary for the computations.

Significant characteristics of two-stroke-cycle engines. - Two types of uniflow two-stroke-cycle engine are schematically shown in figure 1. A comparison of the exhaust-flow areas of these two uniflow two-stroke-cycle engines and a four-stroke-cycle engine is shown in figure 2. These flow areas are corrected for engine displacement and flow coefficients. The flow area of the two-stroke-cycle engine is necessarily greater than that of the four-stroke-cycle engine; however, the larger flow area and faster full-opening of the two-stroke-cycle exhaust valve (about one-fifth the time for the opening of the four-stroke-cycle exhaust valve) probably is generally unappreciated when considering the kinetic energy of the exhaust gases that is available for recovery by a blowdown turbine. The necessary rapid valve opening is, of course, more easily accomplished with the crank-operated sleeve valve than with the cam-operated poppet valve because of the simpler motion of the sleeve valve and because inertia forces at high engine speeds are less of a problem.

The exhaust valve of the two-stroke-cycle engine of necessity opens earlier in the stroke and much more rapidly than the exhaust valve of the four-stroke-cycle engine, which results in some loss in engine power. The combination of earlier opening, which provides exhaust gases at higher initial pressure and temperature, and more rapid opening, which reduces throttling losses, results, however, in exhaust gases of very high kinetic energy during the blowdown period. These gases can advantageously be further expanded to exhaust pressure through the blowdown turbine and then to atmospheric pressure through the steady-flow turbine. The loss in component-engine power caused by early exhaust-valve opening is thus to a great extent recovered by the turbines.

Cylinder charging is somewhat more difficult in the two-stroke-cycle engine than in the four-stroke-cycle engine because of the shorter time available and because the piston motion does not aid in scavenging the cylinder. For this analysis, the inlet valve of the two-stroke-cycle engine is considered to close later than the exhaust valve, thereby permitting some supercharging independent of the exhaust pressure. It is assumed that most of the pressure loss in flow through the cylinder is across the exhaust valve and that the inlet-port area is sufficiently large to accomplish this result. Unlike the four-stroke-cycle engine, which can operate at ratios of exhaust back pressure to inlet-manifold pressure p_e/p_m above 1.0, the two-stroke-cycle engine must operate at values of p_e/p_m less than 1.0 to obtain flow through the cylinder. Because the exhaust pressure available for the steady-flow turbine is limited to values less than inlet pressure, the blowdown turbine becomes more necessary in the two-stroke-cycle than in the four-stroke-cycle composite engine. The blowdown turbine can be designed to impose no increase in engine exhaust pressure, which would allow an increase in net output without penalizing either the component engine or the steady-flow turbine.

An example of the high specific powers that have been obtained from a two-stroke-cycle engine is shown in figure 3. These results were obtained with a $4\frac{5}{8}$- by 7-inch two-stroke-cycle engine (with poppet exhaust valves) using gasoline injection and spark ignition at an effective compression ratio of 6.4. The high specific fuel consumption of this two-stroke-cycle engine was caused by the rich mixture of fuel and air and by the incomplete expansion of the combustion gases due to the early opening of the exhaust valves. The loss in power due to early valve opening would be partly recovered by compounding with exhaust-gas turbines.

Operation of blowdown and steady-flow turbines. - A schematic diagram of the component engine, the compressor, and the turbines considered in this analysis is shown in figure 4. For this analysis it is assumed that the blowdown turbine recovers 60 percent of the energy available for blowdown. Only engine combustion gases produce blowdown-turbine power. The scavenging air leaving the exhaust port is considered to bypass the blading of the blowdown turbine by some mechanical means, but may pass through the shroud of the turbine and aid in cooling. After passing around the blading, the scavenging air is mixed with the combustion gases that have passed through the blowdown turbine. The larger the quantity of scavenging air, the lower will be the temperature of this resultant mixture. This gas mixture at engine exhaust back pressure is available for producing work in the steady-flow turbine by expanding from exhaust back pressure to atmospheric pressure. The exhaust from the steady-flow turbine is available for a small amount of jet propulsion but inasmuch as the results are presented only on a shaft-horsepower basis, the jet energy is omitted from the calculations. Some inlet ram can be produced by the forward speed of the airplane but, for the same reason, consideration of ram-pressure rise and ram drag is also omitted from the analysis.

When the maximum temperature of the mixture entering the steady-flow turbine is to be limited, scavenging air at exhaust pressure is bled from the compressor or furnished by an auxiliary compressor on the same shaft and is mixed with the gases entering the steady-flow turbine.

Method of computation. - The power developed by the two-stroke-cycle component engine was calculated from the rate of combustion-air flow through the engine and from thermal efficiencies of a four-stroke-cycle aircraft engine over a wide range of compression ratios. Use of these efficiencies was necessary because of the lack of similar data for the two-stroke-cycle engine. Unpublished data for a two-stroke-cycle engine, however, were used as a basis for estimating the air flow of the component engine.

For the computation of the net brake horsepower of the composite engine, the compressors and the steady-flow turbine are assumed to be on the same shaft and the difference between the power developed by the steady-flow turbine and that required by the compressor is assumed to be transmitted through gears to the engine crankshaft with a gear efficiency of 90 percent. An efficiency of 95 percent is assumed for the gearing used to transmit power from the blowdown turbine to the crankshaft. The net power output is defined as the sum of the various quantities of power delivered to the crankshaft.

The general method of analysis is to vary the ratio of exhaust pressure to inlet pressure and to compute the powers developed by the several engine components for the chosen conditions of operation. Neither the power required for intercooling nor that required for engine cooling is considered in the analysis.

The derivations of equations applicable to the proposed cycle are presented in the appendix.

The assumed operating conditions are:

Maximum cylinder pressure, pounds per square inch	1200
Temperature of gas mixture entering steady-flow turbine:	
(a)	Maximum unlimited
(b)	1600° F maximum limit
Piston displacement above inlet ports, cubic feet	1.0
Engine speed, rpm	2000
Compressor efficiency, percent	80
Steady-flow-turbine efficiency, percent	80
Blowdown-turbine efficiency, percent	60
Steady-flow-turbine reduction-gear efficiency, percent	90
Blowdown-turbine reduction-gear efficiency, percent	95
Fuel-air ratio in engine cylinder	0.067
Inlet-manifold air temperature, $^{\circ}$F	200
Altitude, feet	30,000

RESULTS OF ANALYSIS

Operation of the composite engine is considered for the following cases:

1. Maximum available temperature of gas mixture to steady-flow turbine unlimited

2. Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F

3. Maximum available temperature of gas mixture to steady-flow turbine unlimited; blowdown turbine omitted

4. Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F; blowdown turbine omitted

The curves for performance at maximum available temperature of the gas mixture to the steady-flow turbine are included only to show

the maximum possible performance. If the calculated performance for these conditions had been poor, there would have been no point in further analysis. More feasible performance values will be found on the curves for operation at limited gas-mixture temperature.

Effect of p_e/p_m on net power output. - The effect of change in the ratio of exhaust to inlet-manifold pressure p_e/p_m on net brake specific horsepower with maximum available temperature of the gas mixture to the steady-flow turbine unlimited is shown in figure 5 for various compression ratios. For each compression ratio, the trend of the power curves is to increase to a maximum and then decrease as the pressure ratio p_e/p_m is increased. This trend in net power is generally the same as the trend of the difference between turbine and compressor powers because over a large range of p_e/p_m the component-engine power changes relatively little. The changes in net brake specific horsepower caused by changes in the ratio of p_e/p_m or by changes in component-engine compression ratio are due primarily to the shifting of the load between the turbine-compressor unit and the component engine. Maximum power output occurs at a p_e/p_m of approximately 0.9. As shown later, if the component engine had a lower scavenging ratio for a given ratio of p_e/p_m than that of the component engine considered for this calculation, the maximum power output would occur at a lower ratio of p_e/p_m than 0.9.

The effect on the net brake specific horsepower of limiting the maximum temperature of the gases entering the steady-flow turbine to 1600° F and of omitting the blowdown turbine is shown in figure 6 for a compression ratio of 3. Because the component engine considered had a relatively high scavenging ratio for a given value of p_e/p_m, exhaust temperatures are fairly low and temperature limitation therefore has little effect, particularly when the blowdown turbine is included in the system. The general effect of limiting the maximum gas temperature is to decrease the maximum power output. Omission of the blowdown turbine materially reduces the power output throughout the entire operating range.

Effect of p_e/p_m on net specific fuel consumption. - The effect of change in the pressure ratio p_e/p_m on net brake specific fuel consumption with the maximum available temperature of the gas mixture entering the steady-flow turbine unlimited is shown in figure 7 for various compression ratios. Net specific fuel consumption is high for low values of p_e/p_m and decreases as the ratio p_e/p_m is increased. The changes in net specific fuel consumption with changes

in p_e/p_m and compression ratio are due to the shifting of the load between the component engine and the turbine-compressor unit. Over the range of operation chosen, minimum net specific fuel consumption occurs at a p_e/p_m of approximately 0.9. If the component engine had a lower scavenging ratio for a given value of p_e/p_m than that of the component engine considered, the fuel-consumption curves would be flatter.

The effect on net brake specific fuel consumption of temperature limitation and the omission of the blowdown turbine at a compression ratio of 3 is shown in figure 8. Temperature limitation again has little effect on the composite-engine performance; it slightly increases the minimum net brake specific fuel consumption. Omission of the blowdown turbine increases the net brake specific fuel consumption over the entire range of operation.

<u>Effect of compression ratio on performance</u>. - The highest net specific horsepowers occur at the lowest component-engine compression ratios and the lowest net brake specific fuel consumptions occur at the highest compression ratios (figs. 5 and 7, respectively). Highest power outputs occur at the lowest compression ratios because the maximum cylinder pressure in the component engine is kept constant; thus, at the low compression ratios, a high inlet-manifold pressure may be used. The high inlet-manifold pressures, together with increased combustion-chamber volume available at the low compression ratios, increase the mass flow through the system, which results in higher outputs. Lower net specific fuel consumptions occur at the higher compression ratios because the component engine is more efficient and takes a greater share of the load.

For convenience, the curves of net brake specific horsepower have been plotted against net brake specific fuel consumption in figure 9. From this figure the net power and corresponding fuel consumption may be obtained for any desired operating condition.

The most probable method of operation is with the maximum temperature of the gas mixture entering the steady-flow turbine limited to 1600° F (fig. 9(b)). When the engine operates at the optimum value of $p_e/p_m = 0.9$, the net brake specific output at a compression ratio of 3 is 2.62 horsepower per cubic inch of piston displacement with a net brake specific fuel consumption of 0.353 pound per horsepower hour. At a compression ratio of 10, the net brake specific power and fuel consumption are 0.71 horsepower per cubic inch and 0.303 pound per horsepower hour, respectively. Operation at other compression ratios gives performance values between these quantities.

Effect of p_e/p_m and compression ratio on gas-mixture temperature. - The effect of change in p_e/p_m and in compression ratio on the temperature of the gas mixture entering the steady-flow turbine, with and without the blowdown turbine, is shown in figure 10. The gas temperatures are lower with the blowdown turbine because the blowdown turbine absorbs some of the exhaust-gas energy. All temperatures are comparatively low because the large amount of scavenging air passing through the component engine dilutes the combustion gases.

Load distribution. - In order to determine load distribution, the power developed by each component of the composite engine was plotted as a percentage of the maximum power output for compression ratios of 3 and 6 (fig. 11). The lower the compression ratio, the greater is the percentage of net power absorbed by the compressor or delivered by the turbines.

Effect of changes in inlet-manifold temperature, scavenging ratio, and fuel-air ratio. - The effect of changes in inlet-manifold temperature, in scavenging ratio, and in fuel-air ratio on net brake specific horsepower and net brake specific fuel consumption is shown in figures 12 and 13, respectively, for different operating conditions. Also included for comparison is a curve of standard performance at the same compression ratio (in this case, 6). The same inlet-manifold pressures used in the standard curve are used in the curves showing the effects of the various changes; maximum cylinder pressures, however, are not necessarily the same. Standard scavenging ratio is that defined by equation (7) in the appendix. Standard inlet-manifold temperature and fuel-air ratio are the assumed values, 200° F and 0.067, respectively. When the fuel-air ratio is increased to 0.081, the fuel not burned in the cylinder is assumed to burn in the duct leading to the steady-flow turbine. Sufficient air is available at that point to allow complete combustion.

The curves show that, at a p_e/p_m of 0.9, increasing the inlet-manifold temperature to 300° F with the blowdown turbine included results in a 13-percent decrease (compared with the standard curve) in net brake specific horsepower (fig. 12(a)) but in only a 1-percent increase in net brake specific fuel consumption (fig. 13(a)). With the blowdown turbine omitted, the power reduction (compared with the standard curve with blowdown turbine omitted) is approximately 13 percent (fig. 12(c)), but the increase in fuel consumption is negligible (fig. 13(c)).

Decreasing the fuel-air ratio to 0.054 with the blowdown turbine results in a 10-percent reduction in net brake specific horsepower and a 10-percent reduction in net brake specific fuel consumption at a p_e/p_m of 0.9. With the blowdown turbine omitted, the reduction in power (compared with the standard curve with blowdown turbine omitted) is approximately 8 percent and the decrease in fuel consumption approximately 12 percent.

Increasing the fuel-air ratio to 0.081 with the blowdown turbine results in a small (4-percent) increase in brake specific horsepower at the high ratios of p_e/p_m and a small decrease in power at the low ratios of p_e/p_m. Net brake specific fuel consumption is increased approximately 16 percent at a p_e/p_m of 0.9. With the blowdown turbine omitted, the power is increased (compared with the standard curve with blowdown turbine omitted) approximately 9 percent at a p_e/p_m of 0.9 but at a p_e/p_m of 0.2 the increase is negligible; the increase in specific fuel consumption is approximately 11 percent over most of the range.

The scavenging ratio was assumed to be decreased to a value equal to 70 percent of the standard scavenging ratio. Such a change could be effected by a change in valve timing or in engine speed. Engine speed is unchanged in this analysis. The net brake specific horsepower (fig. 12(a)) is higher than the value for standard conditions at low values of p_e/p_m and lower at high values of p_e/p_m. The net brake specific fuel consumption (fig. 13(a)) is lower than the value for standard conditions for all ratios of p_e/p_m up to about 0.8. Whereas for all the other operating conditions investigated the maximum power occurs at a p_e/p_m of 0.9, maximum power occurs at a p_e/p_m of approximately 0.6 when the scavenging ratio is reduced to 70 percent of the standard (fig. 12(a)). The results indicate that, where exhaust temperature is not the limiting factor, excess air above the minimum required for adequate scavenging lowers the efficiency but increases the power output.

SUMMARY OF RESULTS

A theoretical analysis of the performance of a composite engine (uniflow two-sroke-cycle spark-ignition engine, compressor, blowdown turbine, and steady-flow turbine) with operation assumed at a constant maximum cylinder pressure of 1200 pounds per square inch, an engine speed of 2000 rpm, an inlet-manifold temperature of 200° F, steady-flow-turbine and compressor efficiencies of 80 percent, and an altitude of 30,000 feet, gave the following results:

The highest net specific horsepowers were obtained at the lowest component-engine compression ratios; the lowest net brake specific fuel consumption was obtained at the highest compression ratios. For the component engine considered, the maximum specific power output and minimum specific fuel consumption occurred at a ratio of exhaust to manifold pressure p_e/p_m of approximately 0.9. At a compression ratio of 3 and a ratio of p_e/p_m of 0.9 with a blowdown turbine and with the maximum exhaust temperature limited to 1600° F, the net brake specific power output was 2.62 horsepower per cubic inch of piston displacement and the net brake specific fuel consumption was 0.353 pound per horsepower hour. At a compression ratio of 10 and a p_e/p_m of 0.9, the net brake specific power and fuel consumption were 0.71 horsepower per cubic inch and 0.303 pound per brake horsepower hour, respectively. Exhaust-temperature limitation and omission of the blowdown turbine adversely affected these values. Where exhaust temperature is not the limiting factor, excess air above the minimum required for adequate scavenging lowered the efficiency but did increase the power output.

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APPENDIX - DERIVATION OF EQUATIONS NECESSARY TO ANALYSIS

Abbreviations

The following abbreviations are used in the derivations:

achp horsepower required to drive auxiliary compressor

bhp brake horsepower of component engine

bthp horsepower delivered by blowdown turbine

chp horsepower required to drive compressor

ihp indicated horsepower of component engine (based on power loop)

isfc indicated specific fuel consumption

nbhp net brake horsepower of composite engine (component engine, turbines, and compressor or compressors)

nbsfc net brake specific fuel consumption of composite engine

nbshp net brake specific horsepower of composite engine

sfhp horsepower delivered by steady-flow turbine

Symbols

The following symbols are used in the derivations:

C concentration

$c_{p,a}$ specific heat at constant pressure of entering scavenging air, 0.243 Btu/(lb)($^{\circ}$R)

$c_{p,b}$ specific heat at constant pressure of combustion gases during blowdown process, 0.3 Btu/(lb)($^{\circ}$R)

$c_{p,e}$ specific heat at constant pressure of gas mixture entering steady-flow turbine, 0.3 Btu/(lb)($^{\circ}$R)

$c_{p,g}$ specific heat at constant pressure of gas mixture before addition of auxiliary air, 0.3 Btu/(lb)($^{\circ}$R)

$c_{v,s}$	specific heat at constant volume of gases in cylinder during scavenging process, 0.243 Btu/(lb)(°R)
F/A	fuel-air ratio in cylinder of component engine
N	speed of component engine, rpm
p_a	ambient-air pressure at altitude of 30,000 feet, 8.88 in. Hg absolute
p_e	exhaust pressure of component engine, in. Hg absolute
p_m	inlet-manifold pressure of component engine, in. Hg absolute
p_r	exhaust release pressure of component engine, in. Hg absolute
r_c	compression ratio of component engine based on swept volume above inlet ports
r_e	expansion ratio of component engine, 0.80 r_c
R	gas constant during blowdown process, 53.9 ft-lb/(lb)(°R)
T_a	ambient-air temperature at altitude of 30,000 feet, 412° R
T_{ac}	temperature of gases at outlet of auxiliary compressor °R
T_e	temperature of gas mixture entering steady-flow turbine, °R
$T_{e,1}$	temperature of gas mixture entering steady-flow turbine with blowdown turbine included, °R
$T_{e,2}$	temperature of gas mixture entering steady-flow turbine with blowdown turbine omitted, °R
T_f	temperature of gases in cylinder at start of compression after evaporative cooling by fuel, °R
T_m	temperature of air entering inlet manifold of component engine, °R
T_s	temperature of gases in cylinder at completion of scavenging process, °R

T_x	temperature of gases in cylinder at start of scavenging process, °R
v	displacement of component-engine piston above inlet ports, 1 cu ft
v_t	total piston displacement of component engine, cu in.
W_a	weight flow of total air admitted through inlet ports, lb/sec
W_{ac}	weight flow of air through auxiliary compressor, lb/sec
W_c	weight flow of air available for combustion, lb/sec
W_e	weight flow of gases entering steady-flow turbine $(W_t + W_{ac})$, lb/sec
W_f	weight flow of fuel admitted to engine, lb/sec
W_t	weight of air admitted through inlet ports plus weight of fuel, $(W_a + W_f)$, lb/sec
γ_b	adiabatic exponent for combustion gases during blowdown process, 1.30
γ_e	adiabatic exponent for gas mixture entering steady-flow turbine, 1.30
η_b	efficiency of blowdown turbine, 0.60
η_c	adiabatic efficiency of compressors, 0.80
$\eta_{g,b}$	efficiency of reduction gearing between blowdown turbine and component engine, 0.95
$\eta_{g,e}$	efficiency of reduction gearing between steady-flow turbine and compressor and component engine, 0.90
η_r	scavenging ratio, ratio of weight of fresh charge delivered through inlet ports to weight of fresh charge that would completely fill cylinder at inlet-manifold density at time of inlet-port closing

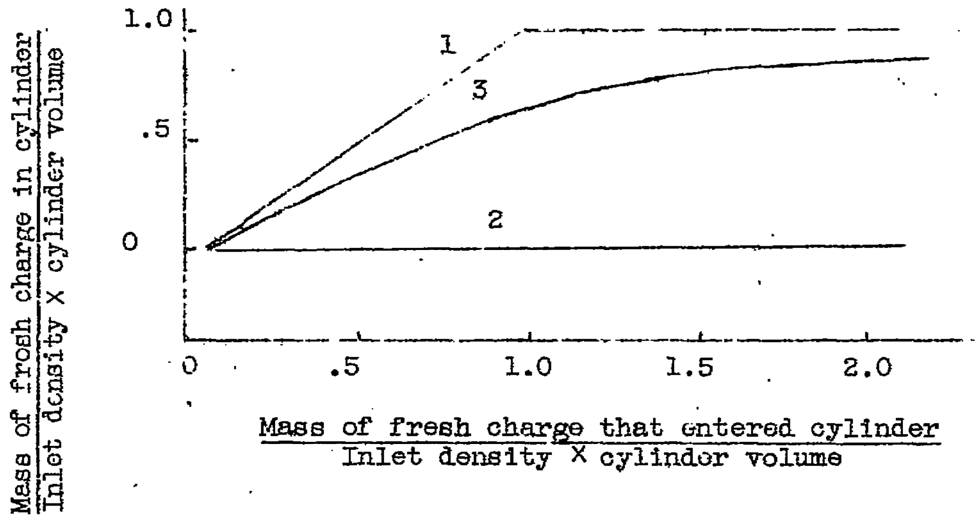
η_s scavenging efficiency, ratio of weight of fresh charge contained in cylinder at time of inlet-port closing to weight of fresh charge that would completely fill cylinder at inlet-manifold density at time of inlet-port closing

η_t adiabatic efficiency of steady-flow turbine, 0.80

ρ_m inlet-manifold density of component engine, lb/cu ft

Equations

Scavenging efficiency. - The three types of scavinging process are: (1) complete scavenging, in which each volume of fresh charge entering the engine cylinder displaces an equal volume of combustion gases, (2) zero scavenging, in which all fresh charge entering is so short-circuited through the exhaust ports that no combustion gases are displaced, and (3) scavenging by perfect mixing, in which each volume of fresh charge entering the cylinder mixes perfectly with the combustion gases and a volume of this mixture, which, an energy balance shows, is sufficient to keep constant the total enthalpy of the contents of the cylinder, is forced out the exhaust ports. The actual scavenging process lies somewhere within the limiting processes and is influenced by the design of the cylinder. These three types of scavenging process are shown in the following sketch:



The equations for curves 1 and 2 are obvious. The equation for curve 3 may be derived as follows:

Consider a cylinder containing a gas B with a volume V at temperature T_B. Gas A at temperature T_A is introduced into this cylinder at such a rate that mixing of the two gases is complete at all times. During this process, which occurs at constant pressure P_A, the mixture of gases is discharged through an exhaust port at such a rate that the total enthalpy of the contents of the cylinder remains constant; that is, the product of the mass of gas in the cylinder and its temperature remains constant.

Let

M mass of gas mixture in cylinder

T temperature of gas mixture in cylinder

M_F mass of gas A in cylinder

M_A mass of gas A introduced into cylinder

ΔM_L mass of gas mixture leaving cylinder when ΔM_A is introduced

C concentration of gas A in cylinder, M_F/M

If ΔM_A enters the cylinder, adding to the cylinder the product $T_A \Delta M_A$, such a mass of gas must leave the cylinder that

$$T_A \Delta M_A = T \Delta M_L$$

and

$$\Delta M_L = \frac{T_A}{T} \Delta M_A$$

The increase in amount of gas A in the cylinder with an amount ΔM_A added to the cylinder and an amount of mixture ΔM_L leaving is

$$\Delta M_F = \Delta M_A - C \Delta M_L$$

or

$$dM_F = dM_A - C\left(\frac{T_A}{T}\right) dM_A$$

$$dM_F = dM_A \left(1 - C \frac{T_A}{T}\right)$$

$$dM_F = dM_A \left(1 - \frac{M_F}{M} \frac{T_A}{T}\right)$$

and, if the subscript B represents the conditions in the cylinder before the scavenging process starts, then

$$dM_F = dM_A \left(1 - \frac{M_F\ T_A}{M_B\ T_B}\right)$$

The solution to this differential equation is

$$M_F = \frac{M_B\ T_B}{T_A}\left(1 - e^{-\frac{M_A\ T_A}{M_B\ T_B}}\right)$$

The constant of integration is determined from the fact that $M_F = 0$ when $M_A = 0$. Now

$$M_B T_B = \frac{P_A V}{R}$$

so that

$$\frac{M_F T_A R}{P_A V} = \left(1 - e^{-\frac{M_A T_A R}{P_A V}}\right)$$

but

$$\frac{M_F T_A R}{P_A V} = \frac{M_F}{\rho_A V} = \eta_s$$

where P_A is the pressure and ρ_A the density of the entering gas, and

$$\frac{M_A T_A R}{P_A V} = \frac{M_A}{\rho_A V} = \eta_r$$

Thus

$$\eta_s = 1 - e^{-\eta_r} \tag{1}$$

The following procedure is used to find the temperature at the end of the scavenging process: The increase of mass of gas in the cylinder is equal to the mass of gas A entering minus the mass of mixture leaving.

Thus

$$dM = dM_A - dM_L$$

or

$$dM = dM_A - \frac{T_A}{T} dM_A$$

and

$$dM = dM_A \left(1 - \frac{T_A}{T} \right)$$

Because

$$M\,T = K$$

where K is a constant, and

$$TdM + MdT = 0$$

and

$$dM = -\frac{MdT}{T} = -\frac{K}{T^2} dT$$

then

$$-\frac{K}{T^2} dT = dM_A \left(1 - \frac{T_A}{T} \right) = dM_A \left(\frac{T - T_A}{T} \right)$$

and

$$\frac{-dT}{T^2} \quad \frac{T}{T - T_A} = \frac{dM_A}{K}$$

The solution to this differential equation, when $K = M_B\,T_B$, is

$$T_s = \frac{T_A}{1 - \left(1 - \frac{T_A}{T_B} \right) e^{-\eta r}} \qquad (2)$$

The constant of integration is determined from the fact that $T = T_B$ when $M_A = 0$.

Unpublished data obtained on a $4\frac{5}{8}$- by 7-inch uniflow two-stroke-cycle engine indicated that its scavenging efficiency was somewhat higher than that predicted by equation (1). For this analysis, the scavenging process was therefore considered to be 75 percent scavenging process 1 and 25 percent scavenging process 3.

Thus between the limits of η_r from 0 to 1.0

$$\eta_s = 0.75\,\eta_r + 0.25\left(1 - e^{-\eta_r}\right) \tag{3}$$

and between the limits of η_r from 1.0 to ∞

$$\eta_s = 0.75 + 0.25\left(1 - e^{-\eta_r}\right) \tag{4}$$

The final scavenging temperature was similarly computed.

Thus between the limits of η_r from 0 to 1.0

$$T_s = 0.75\left[\frac{T_m}{\frac{T_m}{T_x} + \eta_r\left(1 - \frac{T_m}{T_x}\right)}\right] + 0.25\left[\frac{T_m}{1 - \left(1 - \frac{T_m}{T_x}\right)e^{-\eta_r}}\right] \tag{5}$$

and between the limits of η_r from 1.0 to ∞

$$T_s = 0.75\,T_m + 0.25\left[\frac{T_m}{1 - \left(1 - \frac{T_m}{T_x}\right)e^{-\eta_r}}\right] \tag{6}$$

The curve produced from equations (3) and (4) is shown in figure 14. As the curve is prorated between a smooth curve and a curve composed of two straight lines meeting at an angle, the prorated curve at $\eta_r = 1.0$ has a rather sharp curvature, which is smoothed out by fairing the curve in this region.

The value of T_s depends on the value of T_x, which in turn depends largely on the compression ratio of the engine. Over the range of operation and in the region in which points were computed, however, the values of T_s were little affected by changes in compression ratio, and thus an average curve of T_s was chosen and plotted in figure 15. This curve was faired in the region of $\eta_r = 1.0$ in the same manner as the scavenging-efficiency curve.

Scavenging ratio. - If the two-stroke-cycle cylinder is considered as an equivalent orifice, it can be shown that

$$\eta_r = \phi\left[\left(1 - \frac{p_e}{p_m}\right)T_m\right]$$

Data taken on the $4\frac{5}{8}$- by 7-inch two-stroke-cycle engine showed that the scavenging ratio can then be expressed by

$$\eta_r = 0.292 \left[\left(1 - \frac{p_e}{p_m}\right) T_m\right]^{0.341} \tag{7}$$

This curve is plotted in figure 16 together with the corresponding curve of scavenging efficiency cross-plotted from figure 14. Also shown in figure 16 are experimental data points from the $4\frac{5}{8}$- by 7-inch two-stroke-cycle engine that show the excellent agreement between predicted and experimental values.

<u>Inlet-manifold pressure</u>. - In order to illustrate the effect on engine power of changing the compression ratio, some relation must be found between limiting inlet-manifold pressure and compression ratio. Inasmuch as prediction of knock-limited manifold pressure was impractical for the type of component engine considered, no relation between knock-limited manifold pressure and compression ratio could be found. It was therefore decided to limit the inlet-manifold pressure at each compression ratio to a value that would give a maximum cylinder pressure of 1200 pounds per square inch absolute at a fuel-air ratio of 0.067.

Thermodynamic charts of internal-combustion-engine fluids (references 9 and 10) were used in determining the manifold pressure. Addition of fuel is assumed to produce a drop in the temperature of the working fluid of 40° F; so

$$T_f = T_s - 40 \tag{8}$$

The quantity $1 - f$ where f is the fraction of residuals in the cylinder must be known when the thermodynamic charts are used. The quantity $1 - f$ can be obtained from the equation

$$1 - f = \frac{[1 + (F/A)]C}{1 + (F/A)\ C} \tag{9}$$

where

$$C = \eta_s \frac{T_s}{T_m} \tag{10}$$

Curves of inlet-manifold pressure as a function of compression ratio and p_e/p_m are shown in figure 17.

Exhaust release pressure. - The exhaust release pressure was also determined from the thermodynamic charts. A heat loss of 15 percent of the total heat input to the cylinder was assumed to occur at the end of expansion.

Combustion-air weight-flow rate. - From the definition of scavenging efficiency, the combustion-air weight-flow rate is

$$W_c = v \frac{r_c}{r_c - 1} \frac{N\rho_m}{60} \eta_s \tag{11}$$

$$\rho_m = 1.327 \frac{p_m}{T_m} = \frac{1.327 \ p_m}{660} \tag{12}$$

Total-air weight-flow rate. - From the definition of scavenging ratio, total-air weight-flow rate is

$$W_a = v \frac{r_c}{r_c - 1} \frac{N\rho_m}{60} \eta_r \tag{13}$$

where η_r can be calculated from equation (7).

Fuel weight-flow rate. - The fuel weight-flow rate is

$$W_f = W_c \ F/A \tag{14}$$

Indicated horsepower per pound of combustion air per hour (component engine). - Because of the required valve timing in the two-stroke-cycle engine considered, the expansion ratio is less than the compression ratio; in the present case, the expansion ratio is 0.80 of the compression ratio.

It can be shown that for Otto cycles, where the ratio of expansion to compression is in the range of 0.80 to 1.0, the efficiency of the cycle can be based on the expansion ratio with good accuracy.

Because of lack of data on the thermal efficiency of two-stroke-cycle engines at various compression ratios, it is convenient to use four-stroke-cycle data. The close agreement between two-stroke-cycle and four-stroke-cycle thermal efficiencies can be shown by the fact that the indicated specific fuel consumption from the $4\frac{5}{8}$- by 7-inch two-stroke-cycle engine with an expansion ratio of 5.44 was 0.470 pound per indicated horsepower hour, whereas the indicated specific fuel consumption of a conventional four-stroke-cycle engine corrected to

a compression ratio of 5.44 was 0.462 pound per indicated horsepower hour at the same fuel-air ratio.

Data from reference 11 and unpublished data from tests on an air-cooled cylinder were analyzed, and an equation for indicated specific fuel consumption as a function of compression ratio was found to be

$$isfc = \frac{0.1317}{\left[1 - \frac{1}{r_c^{\ 0.22}}\right]} \tag{15}$$

The agreement between values calculated from the equation and experimental results over a range of compression ratios at a fuel-air ratio of 0.067 is shown by the following table:

Compression ratio	Indicated specific fuel consumption (lb/hp-hr)	
	Experimental	Calculated
4.7	0.457	0.457
6.7	.384	.384
7.93	.360	.360
9.68	.335	.335

Equation (15) may be converted to permit calculation of indicated horsepower hour per pound of combustion air. The modified equation may be used for computing the two-stroke-cycle performance by using the expansion ratio instead of the compression ratio. The indicated horsepower hour per pound of combustion air at a fuel-air ratio of 0.067 is then

$$\frac{ihp}{3600\ W_c} = 0.5085 \left(1 - \frac{1}{r_e^{\ 0.22}}\right) \tag{16}$$

Indicated horsepower of component engine. - The indicated horsepower of the component engine is

$$ihp = \frac{ihp}{3600\ W_c} W_c\ 3600 \tag{17}$$

where W_c may be calculated from equation (11) and $\frac{ihp}{3600\ W_c}$ from equation (16).

Listed here in tabular form for comparison are values of indicated specific power calculated according to this analysis and some experimental data from the $4\frac{5}{8}$- by 7-inch two-stroke-cycle engine at a fuel-air ratio of 0.075, an engine speed of 2000 rpm, and an exhaust pressure of 30 inches of mercury absolute:

Manifold pressure (in. Hg abs.)	Specific power (ihp/cu in.)	
	Experimental	Calculated
44	1.047	1.060
46	1.106	1.113
48	1.148	1.163
50	1.173	1.215

Although experimental data are not available for the upper range of power, the table shows that calculated and experimental power outputs agree well in the lower range.

Brake horsepower of component engine. - Data from an investigation conducted for the Army Air Forces on a $5\frac{1}{4}$- by 7-inch uniflow two-stroke-cycle engine showed that pumping power was negligible and that mechanical friction was the largest loss in conversion of indicated horsepower to brake horsepower. Analysis of those data indicated that the friction horsepower of the component engine is approximately 200.

Therefore

$$\text{bhp} = \text{ihp} - 200 \tag{18}$$

Horsepower required by compressor. - The power required to drive the compressor is given by

$$\text{chp} = \left\{ \frac{189.1\, W_a T_a \left[\left(\frac{p_m}{p_a} \right)^{0.283} - 1 \right]}{\eta_c\, 550} \right\} \tag{19}$$

Blowdown-turbine horsepower. - Blowdown in the cylinder is considered to take place from exhaust release pressure to inlet-manifold pressure. The pressure drop across the blowdown turbine is thus from exhaust release pressure to exhaust back pressure at the start of the process, and from inlet-manifold pressure to exhaust back pressure at the end of the process. Scavenging air is considered to bypass the

blowdown-turbine blading but may pass through the shroud of the turbine and assist in cooling the turbine. The scavenging air then mixes with the combustion gases that have passed through the blowdown turbine and the mixture of these gases is available to produce work in the steady-flow turbine.

The horsepower of the blowdown turbine is given by the equation

$$bthp = N \frac{r_e}{r_c} \frac{r_c}{r_c - 1} \frac{778}{33,000} \frac{c_{p,b}}{R\gamma_b} \eta_b \, 0.491 \times 144(p_r - p_m)\left[1 - \gamma_b \, p_e^{\frac{\gamma_b - 1}{\gamma_b}} \frac{\left(p_r^{\frac{1}{\gamma_b}} - p_m^{\frac{1}{\gamma_b}}\right)}{p_r - p_m}\right] \quad (20)$$

The value of η_b was chosen to be 0.60, a conservative assumption based on material from reference 12.

Temperature of mixture entering steady-flow turbine. - An energy balance across the cylinder during the scavenging process results in

$$E_x + H_a - E_s = H_c$$

where

E_x internal energy of gases in cylinder at start of scavenging process, Btu

E_a total enthalpy of entering scavenging air, Btu

E_s internal energy of gases in cylinder at end of scavenging process, Btu

H_c enthalpy of scavenging gases leaving cylinder, Btu

Because

$$E_x = c_{v,s} \, M_x \, T_x$$

$$E_s = c_{v,s} M_s T_s$$

where M is here the weight of the gases and

$$M_s T_s = M_x T_x$$

then

$$E_x = E_s$$

and thus

$$H_a = H_c$$

An energy balance of the blowdown process from the cylinder yields the equation

$$E_r = E_m + w + H_g$$

where

E_r internal energy in cylinder at time of exhaust release, Btu

E_m internal energy in cylinder after blowdown to inlet-manifold pressure, Btu

w work output of blowdown turbine, Btu

H_g enthalpy of combustion gases after leaving blowdown turbine, Btu

The total enthalpy H_e of gases entering the steady-flow turbine is then

$$H_c + H_g = E_r - E_m + H_a - w = H_e$$

$E_r - E_m$ is given by the equation

$$E_r - E_m = \frac{r_e}{r_c} \quad \frac{r_c}{r_c - 1} \quad \frac{c_{p,b}}{R\gamma_b} (p_r - p_m)\ 0.491 \times 144$$

and if the enthalpies are computed from a temperature base of 660° R, the temperature of the exhaust gases entering the steady-flow turbine with blowdown turbine included is

$$T_{e,1} = \frac{\frac{r_e}{r_c - 1}\,\frac{c_{p,b}}{R\gamma_b}\,(p_r - p_m)\;0.491 \times 144 - \frac{bthp\;33{,}000}{N\;778}}{W_t\;c_{p,g}\;\frac{60}{N}} + 660 \qquad (21)$$

The temperature of the gases entering the steady-flow turbine with the blowdown turbine omitted is

$$T_{e,2} = \frac{\frac{r_e}{r_c}\,\frac{r_c}{r_c - 1}\,\frac{c_{p,b}}{R\gamma_b}\,(p_r - p_m)\;0.491 \times 144}{W_t\;c_{p,g}\;\frac{60}{N}} + 660 \qquad (22)$$

Weight-flow rate of auxiliary-compressor air required to reduce temperature of gases entering steady-flow turbine to 1600° F. - The weight-flow rate of auxiliary-compressor air with blowdown turbine included is given by the equation

$$W_{ac} = \frac{W_t\;c_{p,e}\;1400 - W_t\;c_{p,g}\;(T_{e,1} - 660)}{c_{p,a}\;T_{ac} - c_{p,a}\;660 - c_{p,e}\;1400} \qquad (23)$$

The weight-flow rate of auxiliary-compressor air if the blowdown turbine is omitted is

$$W_{ac} = \frac{W_t\;c_{p,e}\;1400 - W_t\;c_{p,g}\;(T_{e,2} - 660)}{c_{p,a}\;T_{ac} - c_{p,a}\;660 - c_{p,e}\;1400} \qquad (24)$$

Horsepower required to drive auxiliary compressor. - The horsepower required to drive the auxiliary compressor is

$$achp = \frac{189.1\;W_{ac}\;T_a\left[\left(\frac{p_e}{p_a}\right)^{0.283} - 1\right]}{\eta_c\;550} \qquad (25)$$

The air temperature at the outlet of the auxiliary compressor is given by

$$T_{ac} = \frac{T_a}{\eta_c}\left[\left(\frac{p_e}{p_a}\right)^{0.283} - 1\right] + T_a \qquad (26)$$

Horsepower of steady-flow turbine. - The horsepower produced by the steady-flow turbine is

$$\text{sfhp} = \frac{778}{550} \eta_t \, c_{p,e} \, W_e \, T_e \left[1 - \left(\frac{p_a}{p_e} \right)^{\frac{\gamma_e - 1}{\gamma_e}} \right] \tag{27}$$

The gas-mixture temperature T_e may be either $T_{e,1}$ if the blow-down turbine is present or $T_{e,2}$ when the blowdown turbine is omitted. When temperature-limited operation is encountered, T_e is 2060° R.

Net brake horsepower. - The net brake horsepower of the composite engine is

$$\text{nbhp} = \text{bhp} + \eta_{g,e} \, (\text{sfhp} - \text{chp} - \text{achp}) + \eta_{g,b} \, (\text{bthp}) \tag{28}$$

If sfhp is less than (chp + achp), $\eta_{g,e}$ is changed to $\frac{1}{\eta_{g,e}}$ and the product $\frac{1}{\eta_{g,e}}$ (sfhp - chp - achp) is algebraically added.

Net brake specific fuel consumption. - The net brake specific fuel consumption of the composite engine is

$$\text{nbsfc} = \frac{W_f}{\text{nbhp}} \, 3600 \tag{29}$$

Net brake specific horsepower. - For convenience, calculations of the component-engine horsepowers and air flows were based on a piston displacement above the ports of 1 cubic foot.

If the ratio of compression ratio based on piston displacement above the inlet ports to the compression ratio based on total piston displacement is taken to be 0.8, a conservative figure, the total piston displacement then becomes

$$v_t = 1728 \times 1.25 \; \frac{r_c - 0.8}{r_c - 1}$$

and the net brake specific power per cubic inch of piston displacement is

$$\text{nbshp} = \frac{\text{nbhp}}{2160} \; \frac{r_c - 1}{r_c - 0.8} \tag{30}$$

At a compression ratio of 6, the piston displacement would be the same as that of a 12-cylinder engine with a bore of 6 inches and a stroke of $6\frac{3}{4}$ inches.

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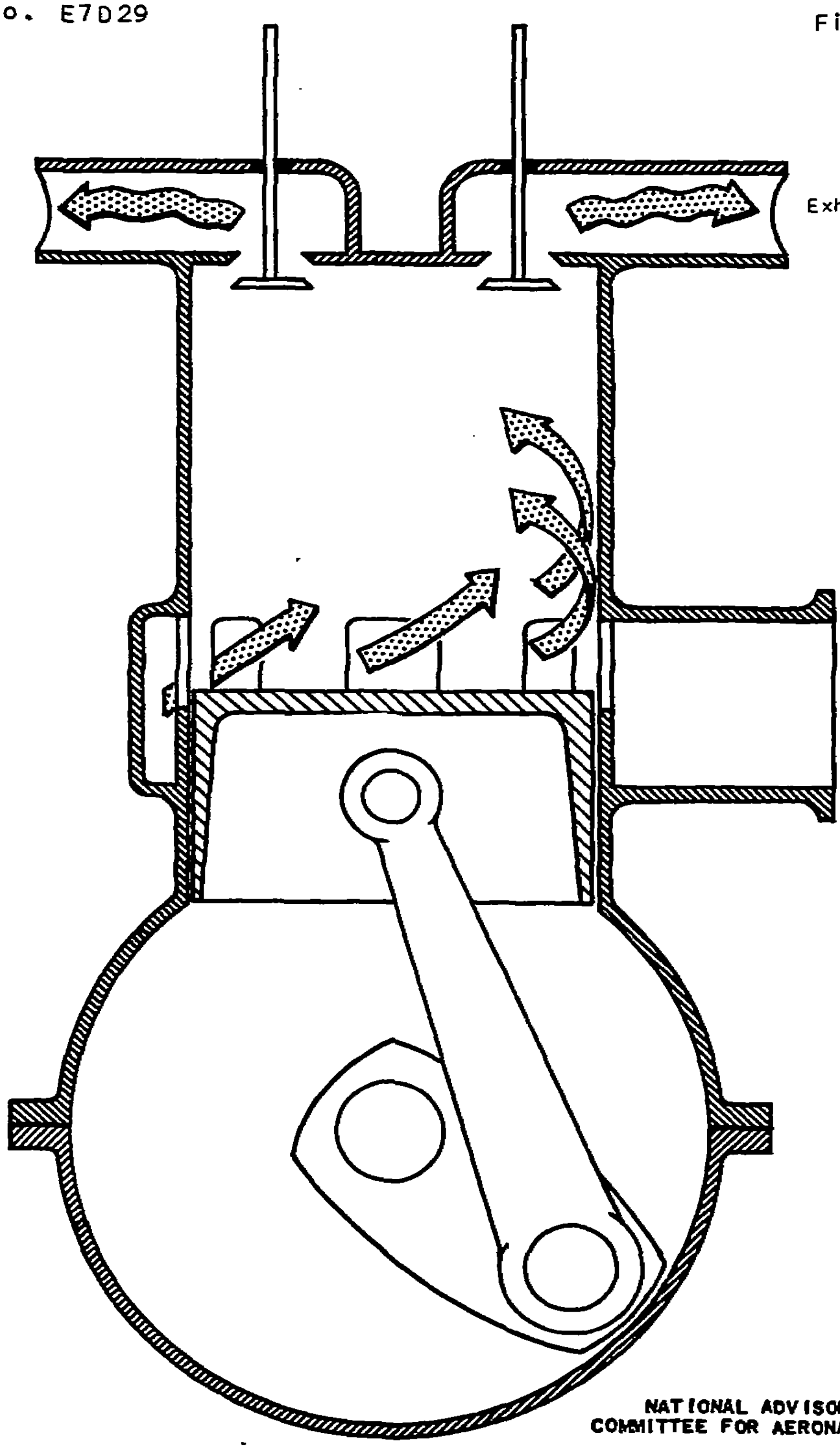
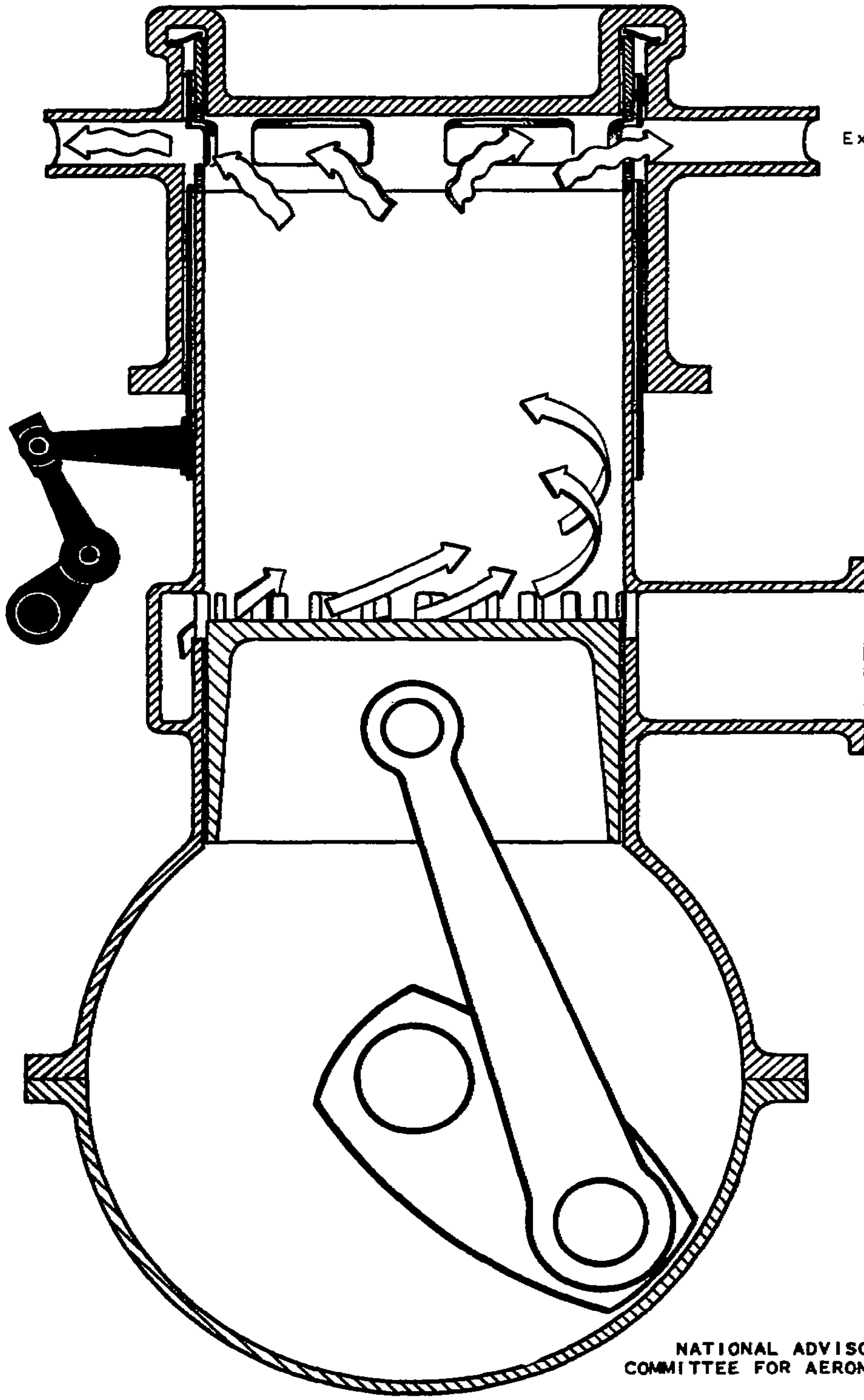


(a) With poppet exhaust valves.

Figure 1. - Schematic diagram of uniflow two-stroke-cycle engine.



(b) With sliding-sleeve exhaust valve.

Figure 1. - Concluded. Schematic diagram of uniflow two-stroke-cycle engine.

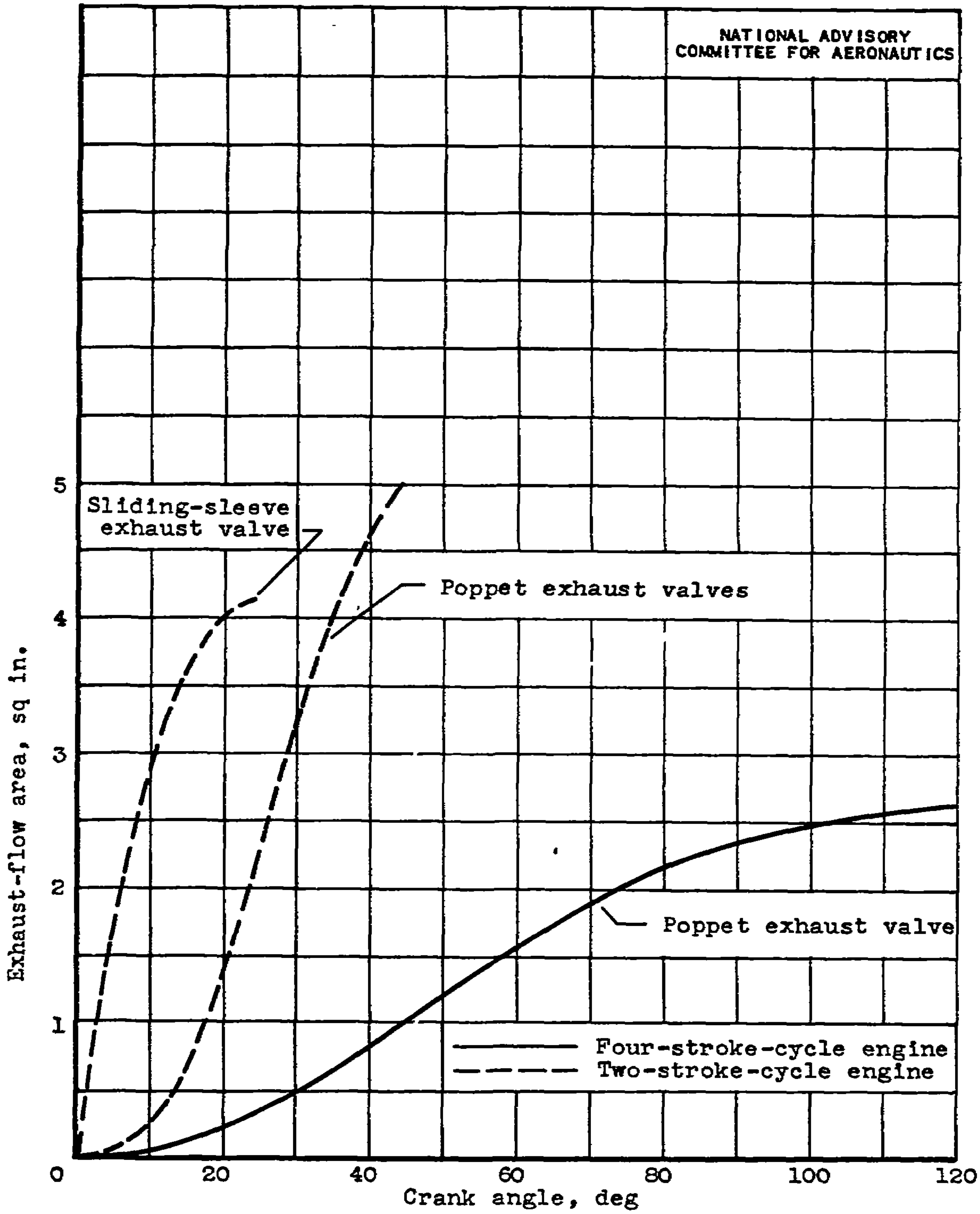


Figure 2. - Comparative exhaust-flow areas for two- and four-stroke-cycle engines from start to full opening.

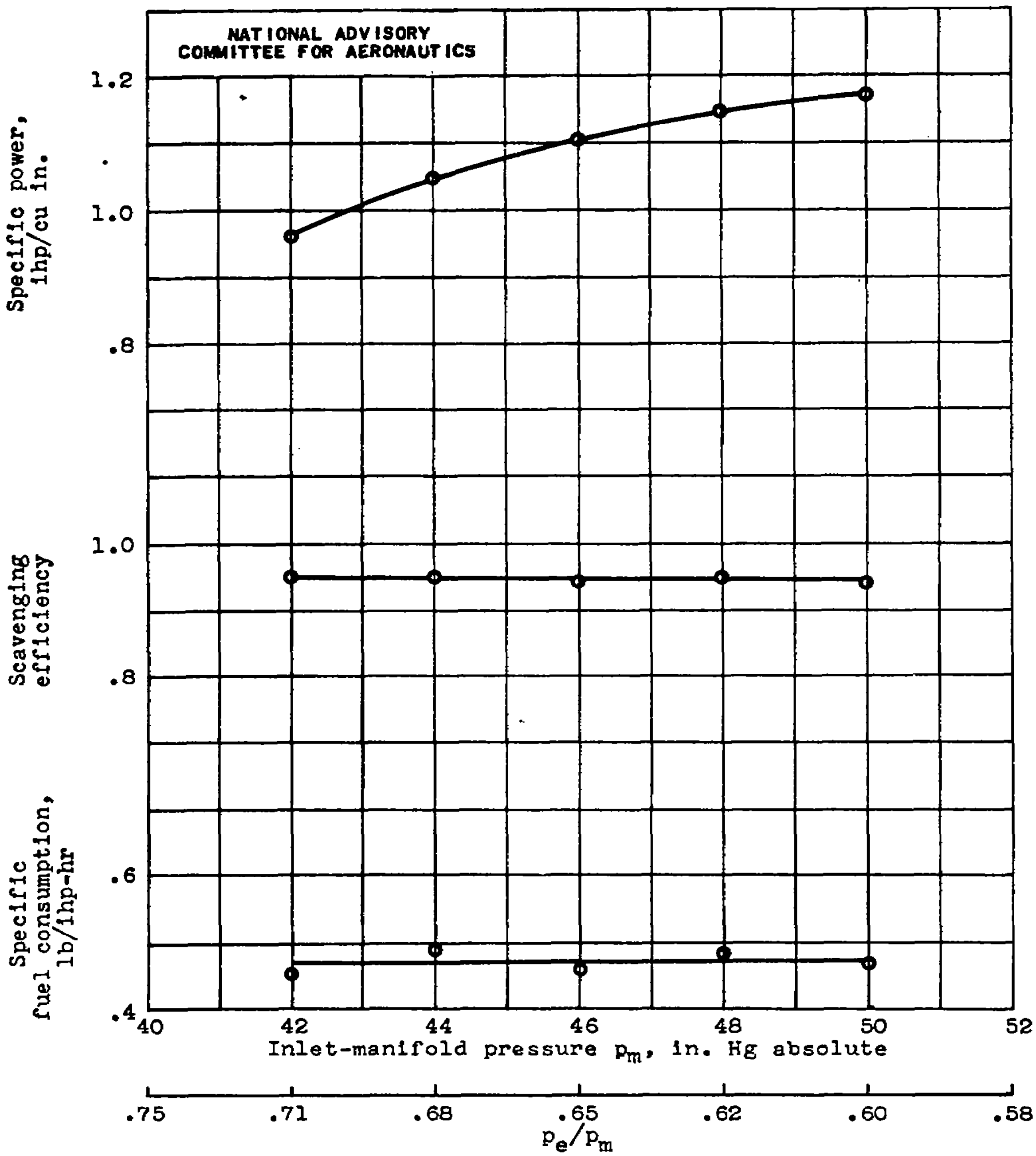


Figure 3. - Performance of $4\frac{5}{8}$- by 7-inch two-stroke-cycle spark-ignition engine with poppet exhaust valves and piston-controlled inlet ports. Atmospheric exhaust; engine speed, 2000 rpm; fuel-air ratio, 0.075; fuel, 100-octane gasoline plus 3 ml TEL per gallon; inlet-air temperature, 92° F; effective compression ratio, 6.4.

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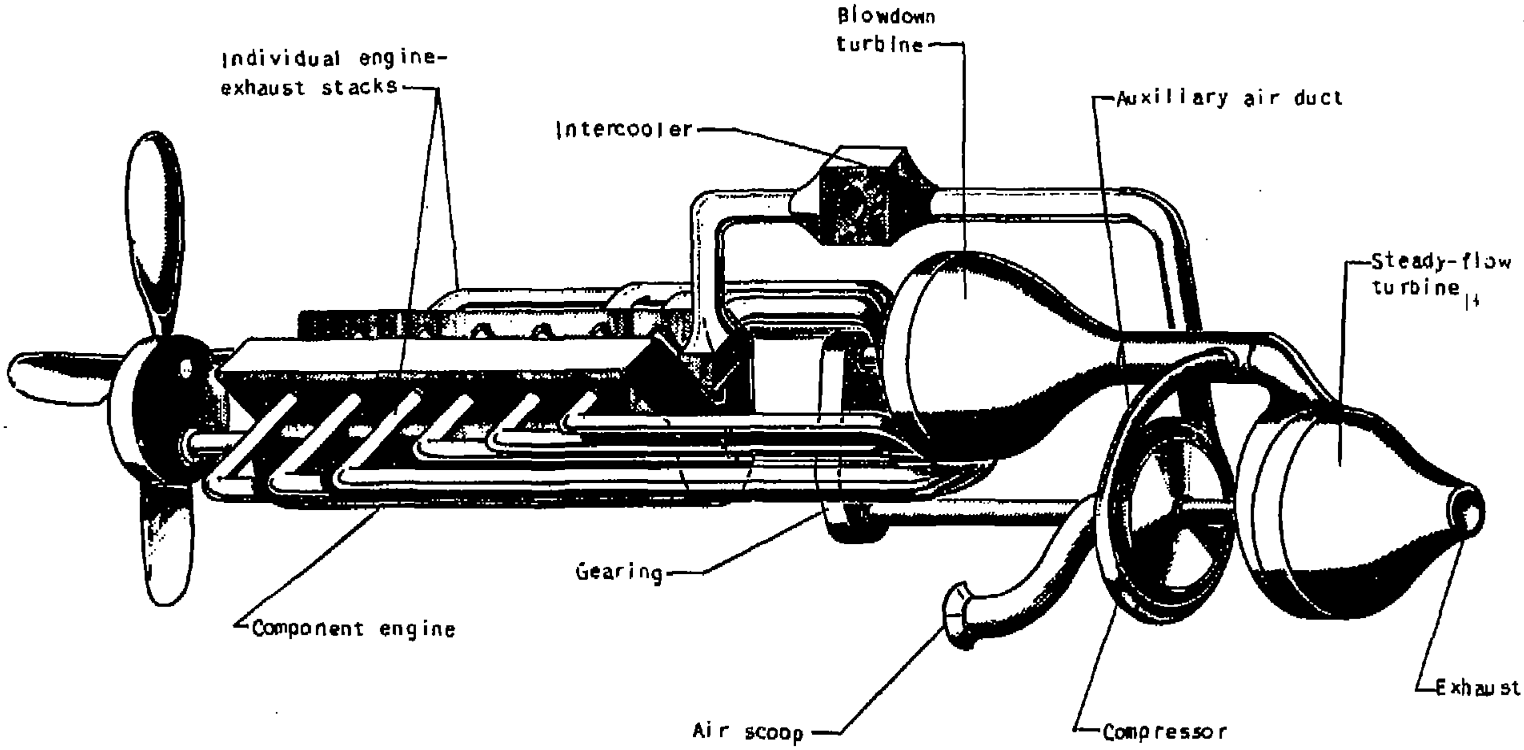

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Figure 4. - Schematic diagram of composite engine consisting of two-stroke-cycle spark-ignition engine, compressor, blowdown turbine, and steady-flow turbine.

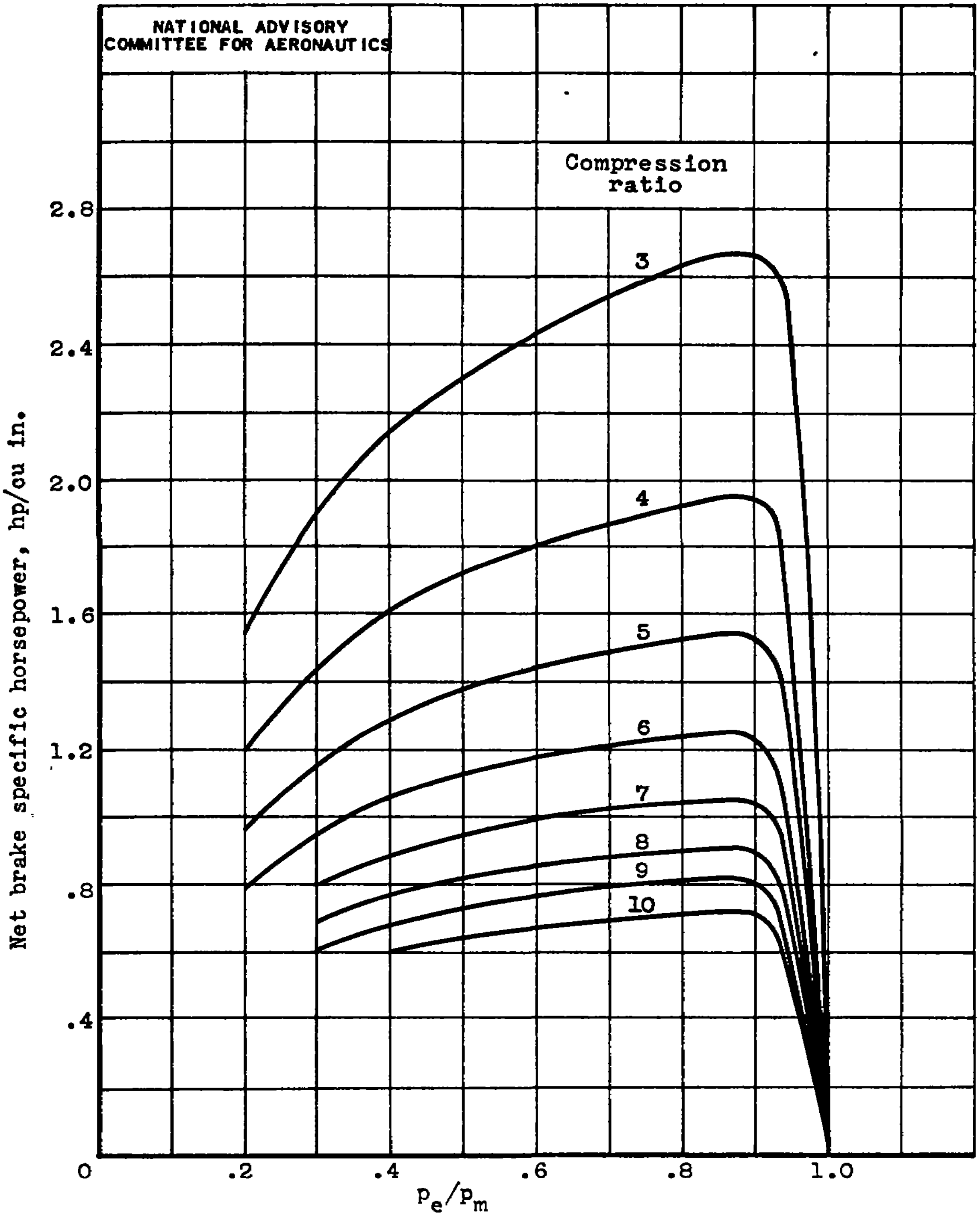

Figure 5. - Effect of pressure ratio p_e/p_m on net brake specific horsepower of composite engine for various compression ratios. Maximum available temperature of gas mixture to steady-flow turbine unlimited.

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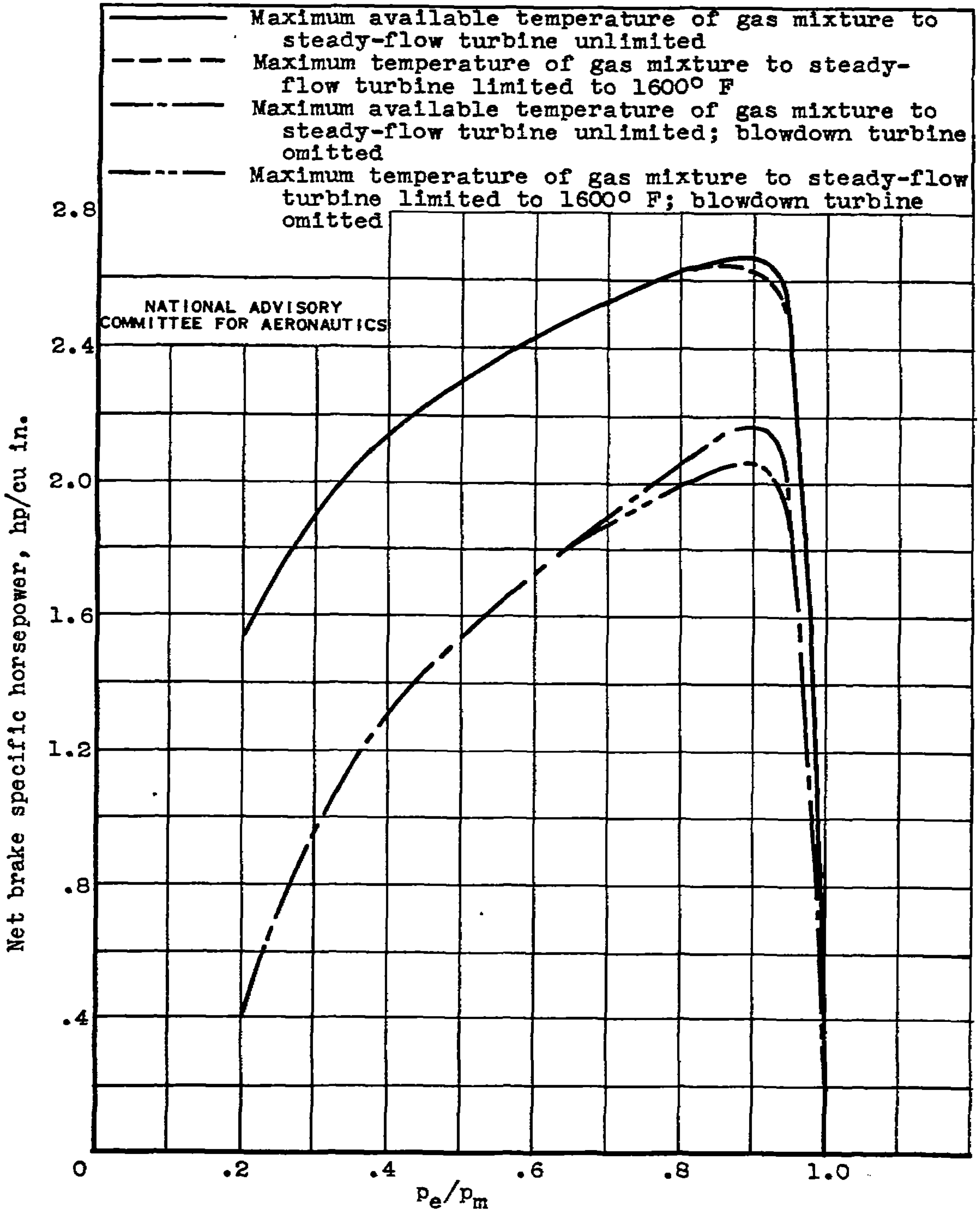


Figure 6. - Effect of pressure ratio p_e/p_m on net brake specific horsepower of composite engine for various methods of operation. Compression ratio, 3.

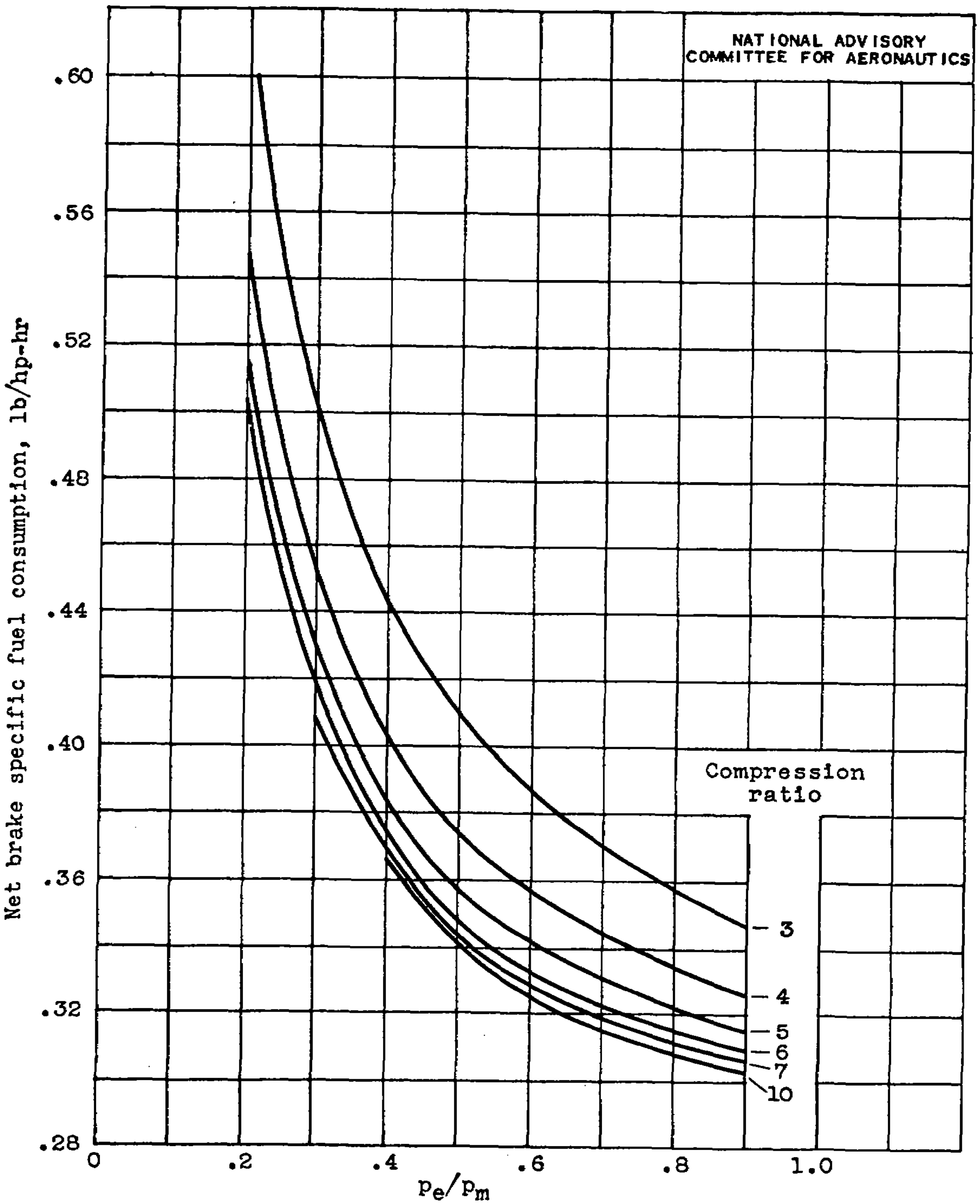


Figure 7. - Effect of pressure ratio p_e/p_m on net brake specific fuel consumption of composite engine for various compression ratios. Maximum available temperature of gas mixture to steady-flow turbine unlimited.

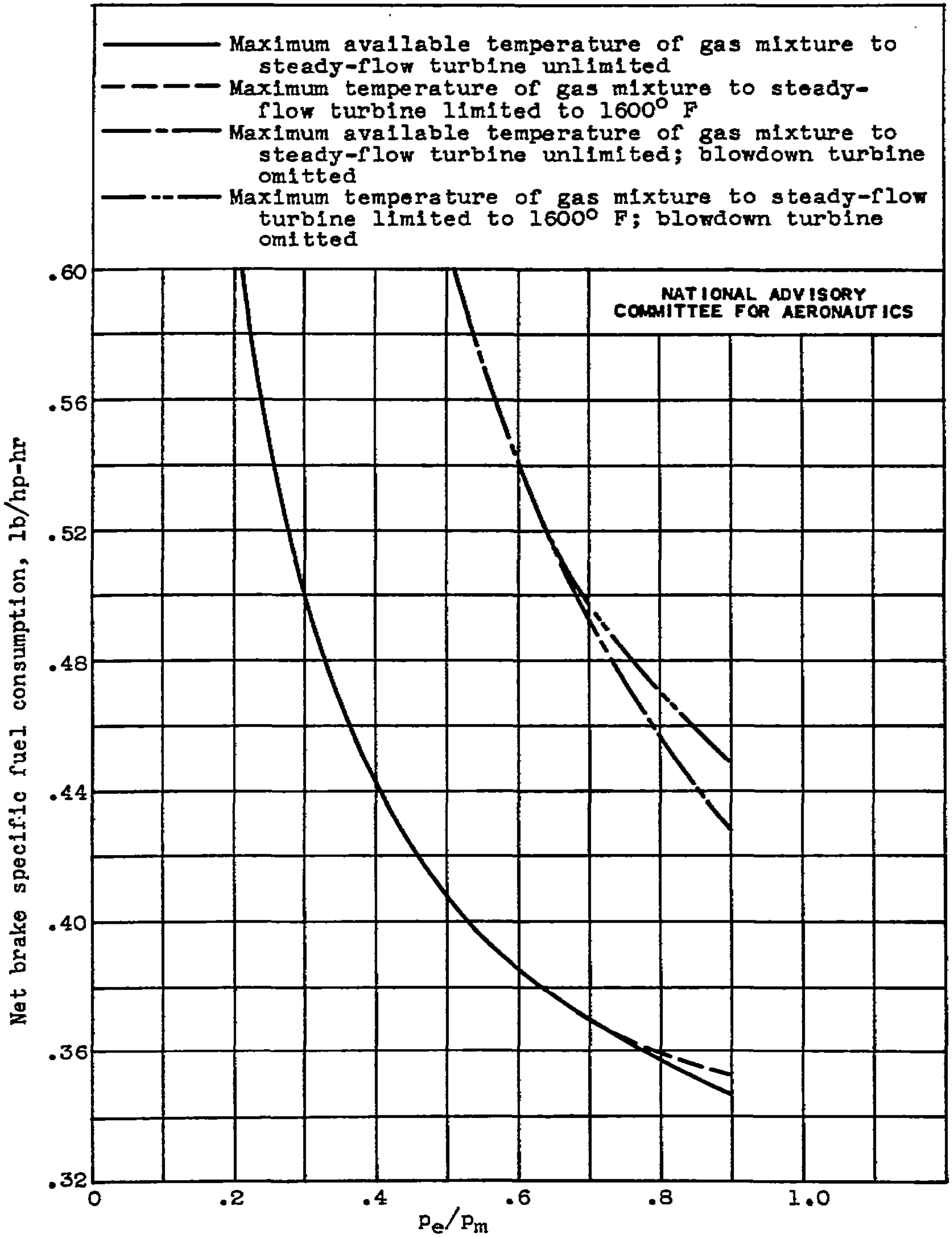
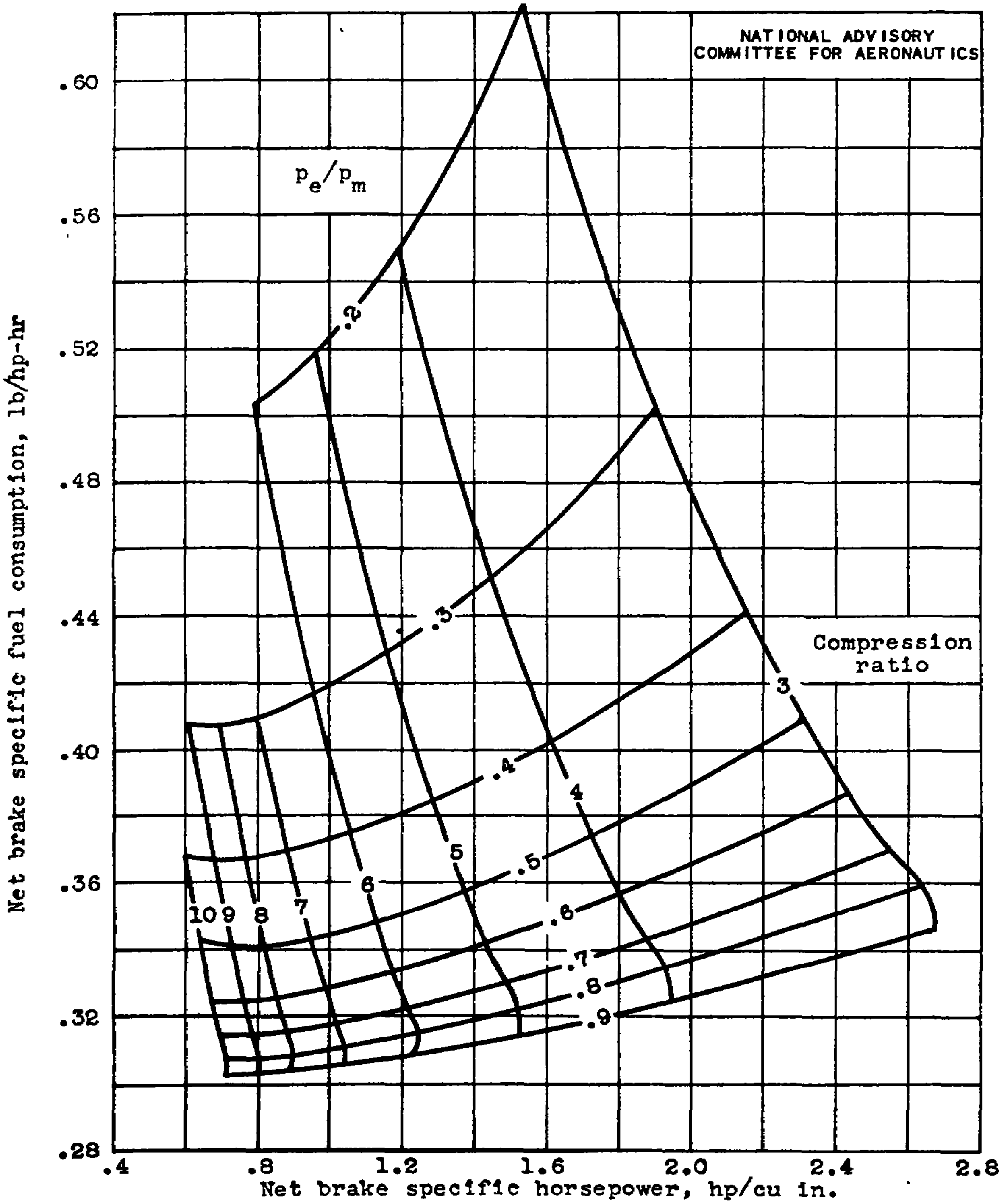


Figure 8. - Effect of pressure ratio p_e/p_m on net brake specific fuel consumption of composite engine for various methods of operation. Compression ratio, 3.



(a) Maximum available temperature of gas mixture to steady-flow turbine unlimited. (Cross-plotted from figs. 5 and 7.)

Figure 9. - Net brake specific horsepower and net brake specific fuel consumption of composite engine.

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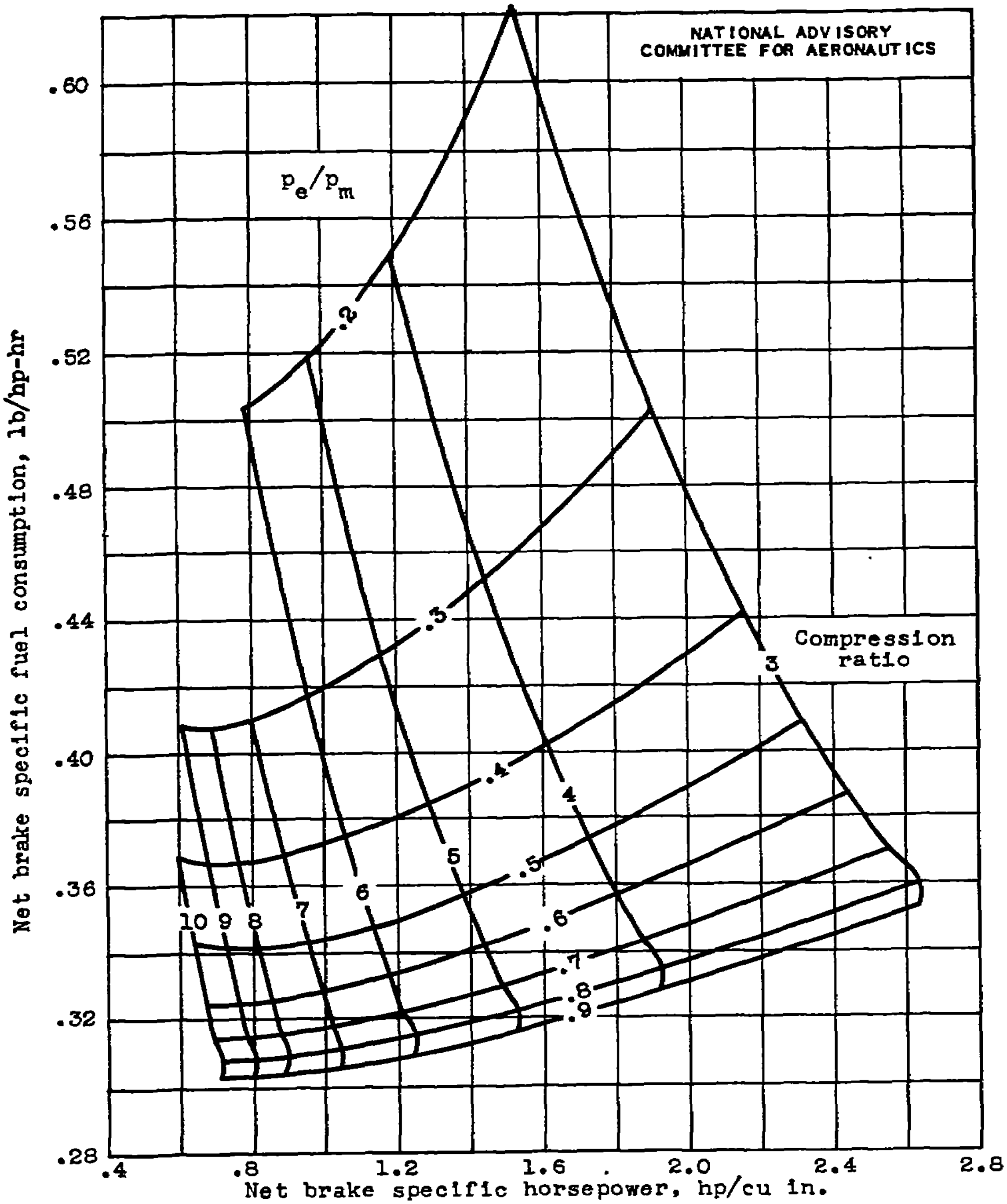
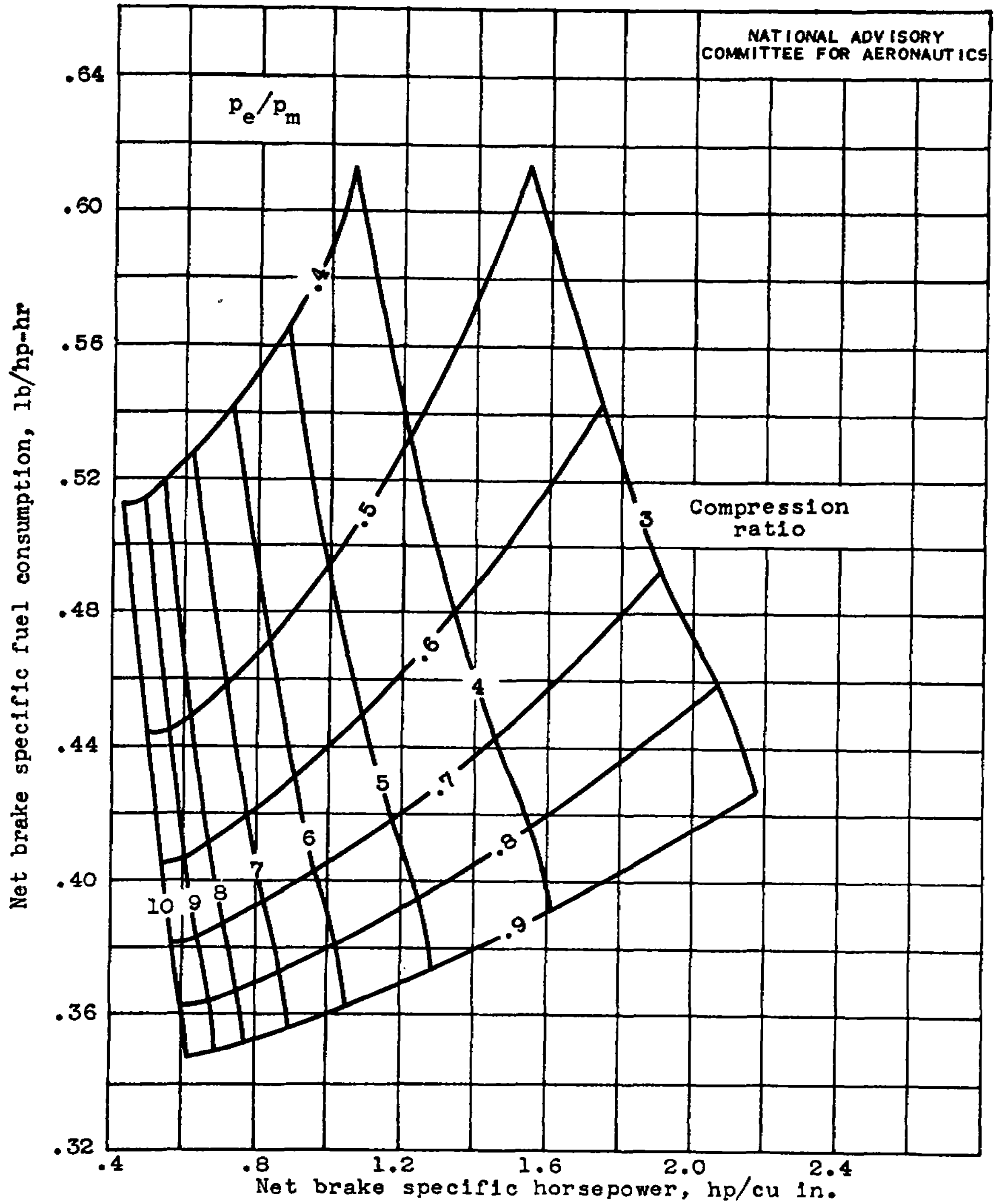


(b) Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F.

Figure 9. - Continued. Net brake specific horsepower and net brake specific fuel consumption of composite engine.



(c) Maximum available temperature of gas mixture to steady-flow turbine unlimited; blowdown turbine omitted.

Figure 9. - Continued. Net brake specific horsepower and net brake specific fuel consumption of composite engine.

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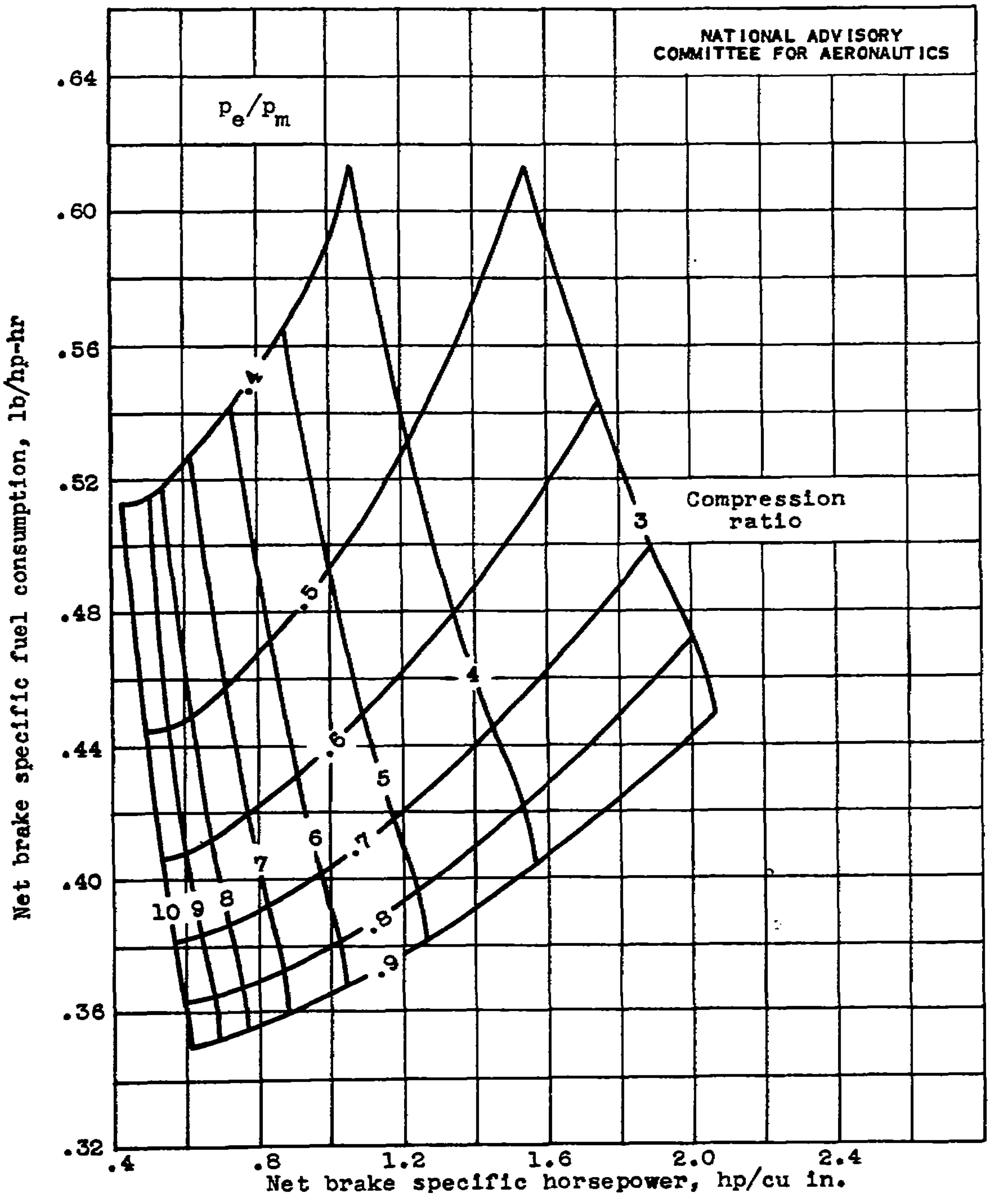


(d) Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F; blowdown turbine omitted.
Figure 9. - Concluded. Net brake specific horsepower and net brake specific fuel consumption of composite engine.

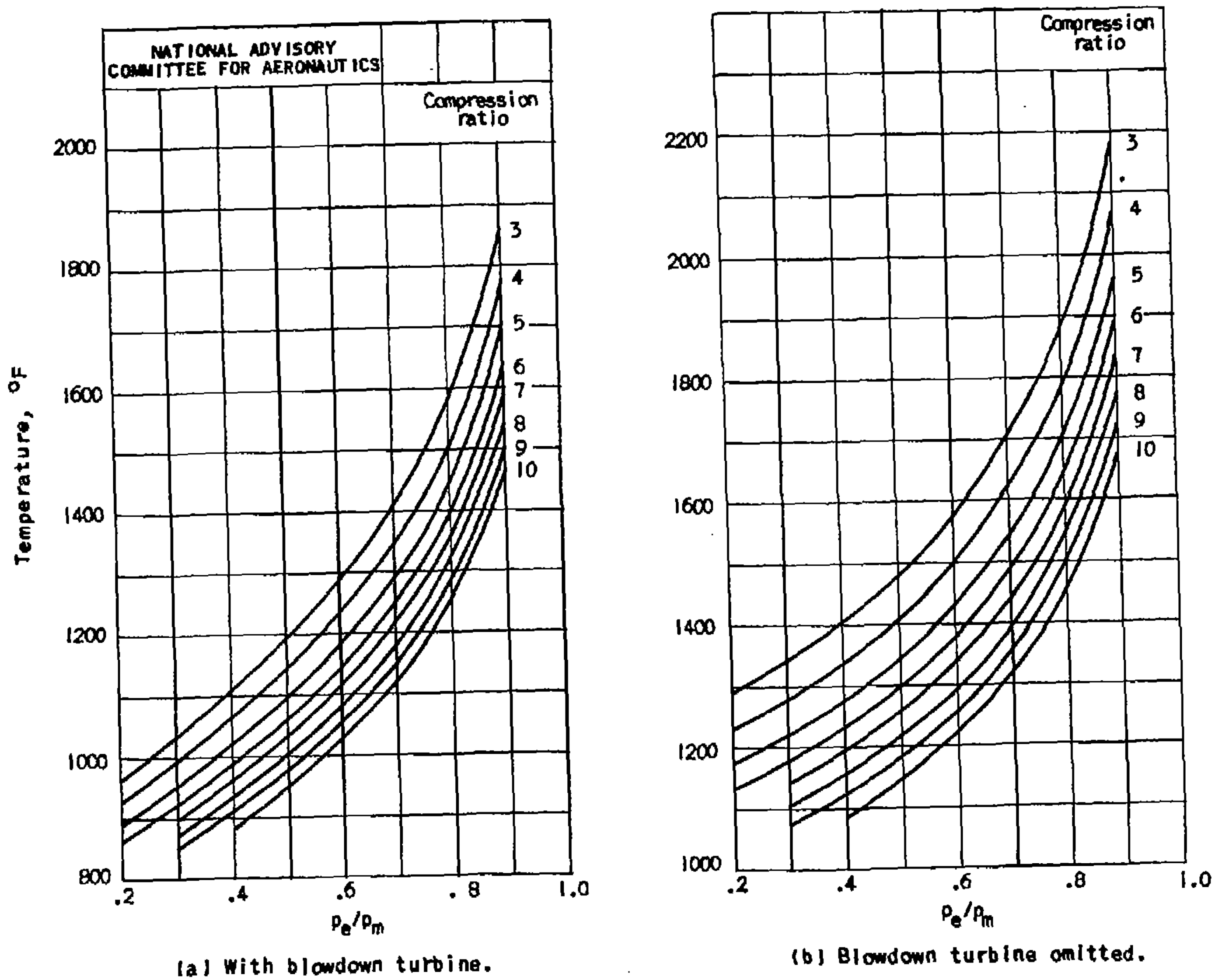
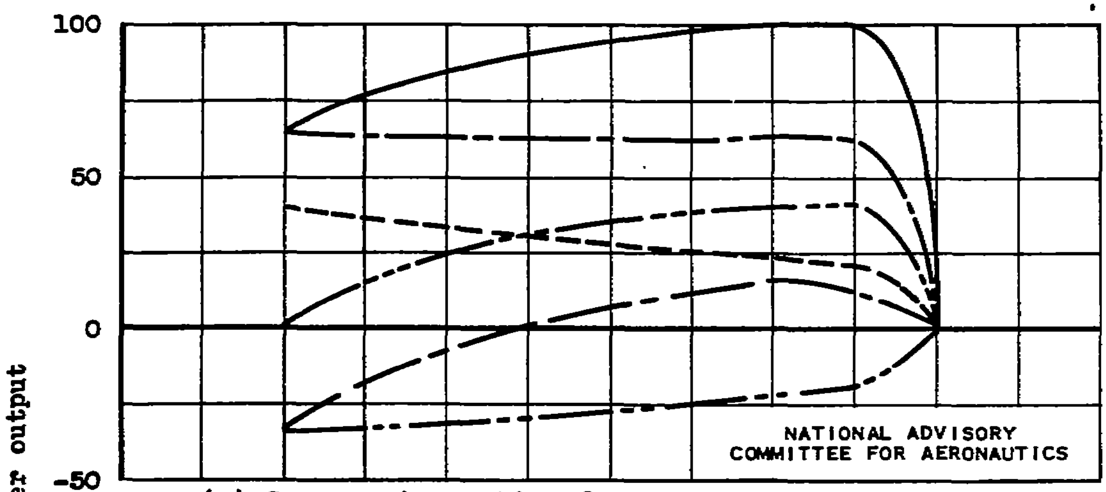
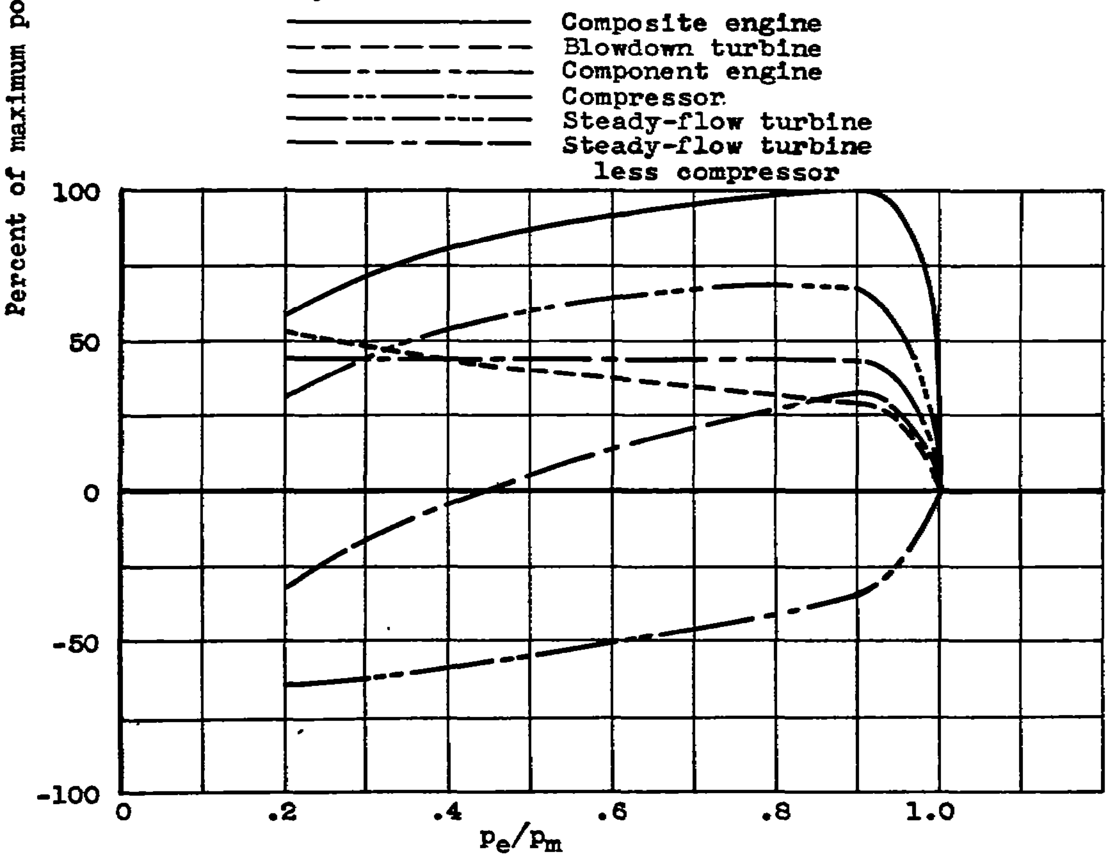


Figure 10. - Effect of pressure ratio p_e/p_m on temperature of gas mixture to steady-flow turbine for various compression ratios.



(a) Compression ratio, 6.



(b) Compression ratio, 3.

Figure 11. - Effect of pressure ratio p_e/p_m and compression ratio on load distribution. Maximum available temperature of gas mixture to steady-flow turbine unlimited.

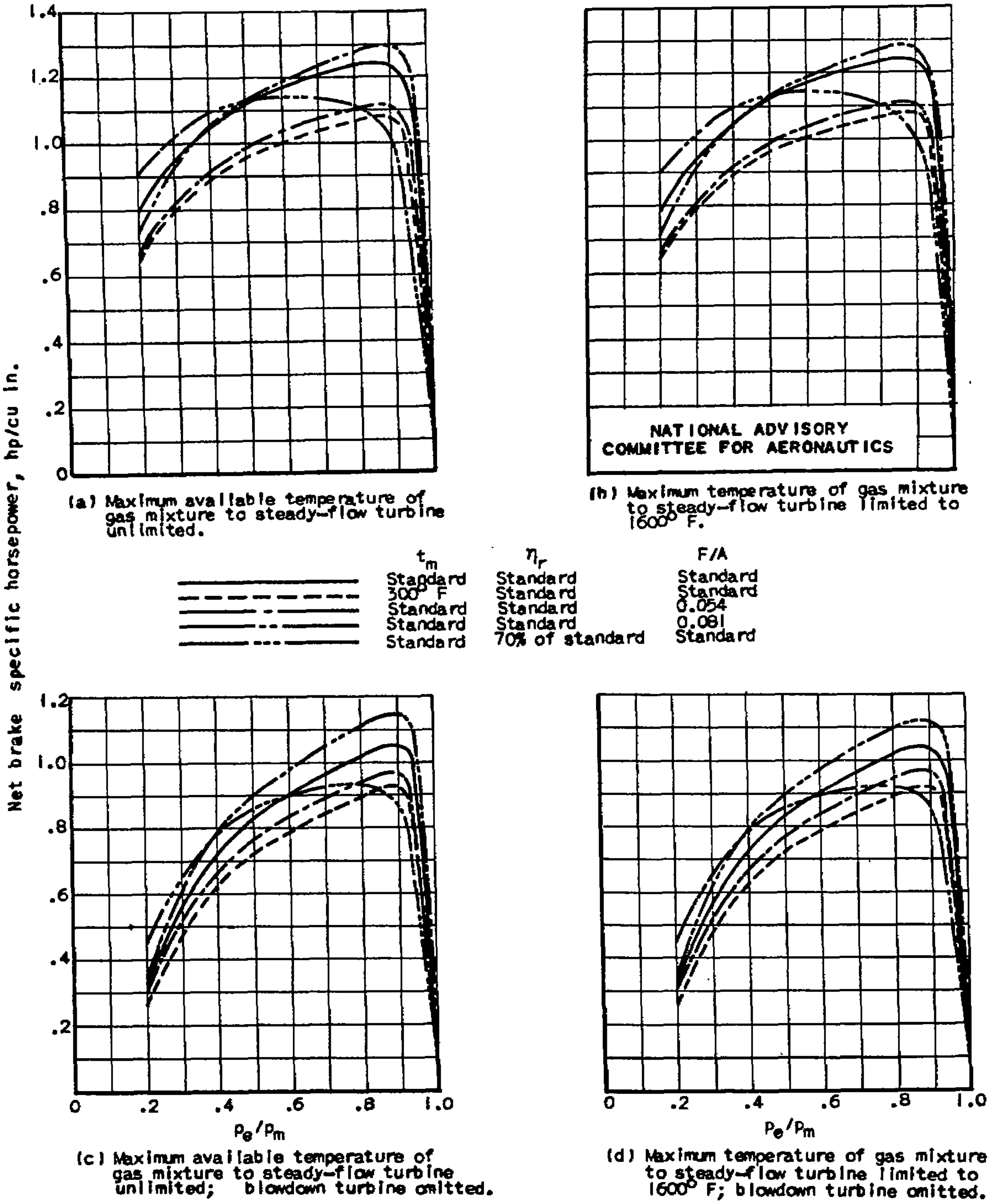
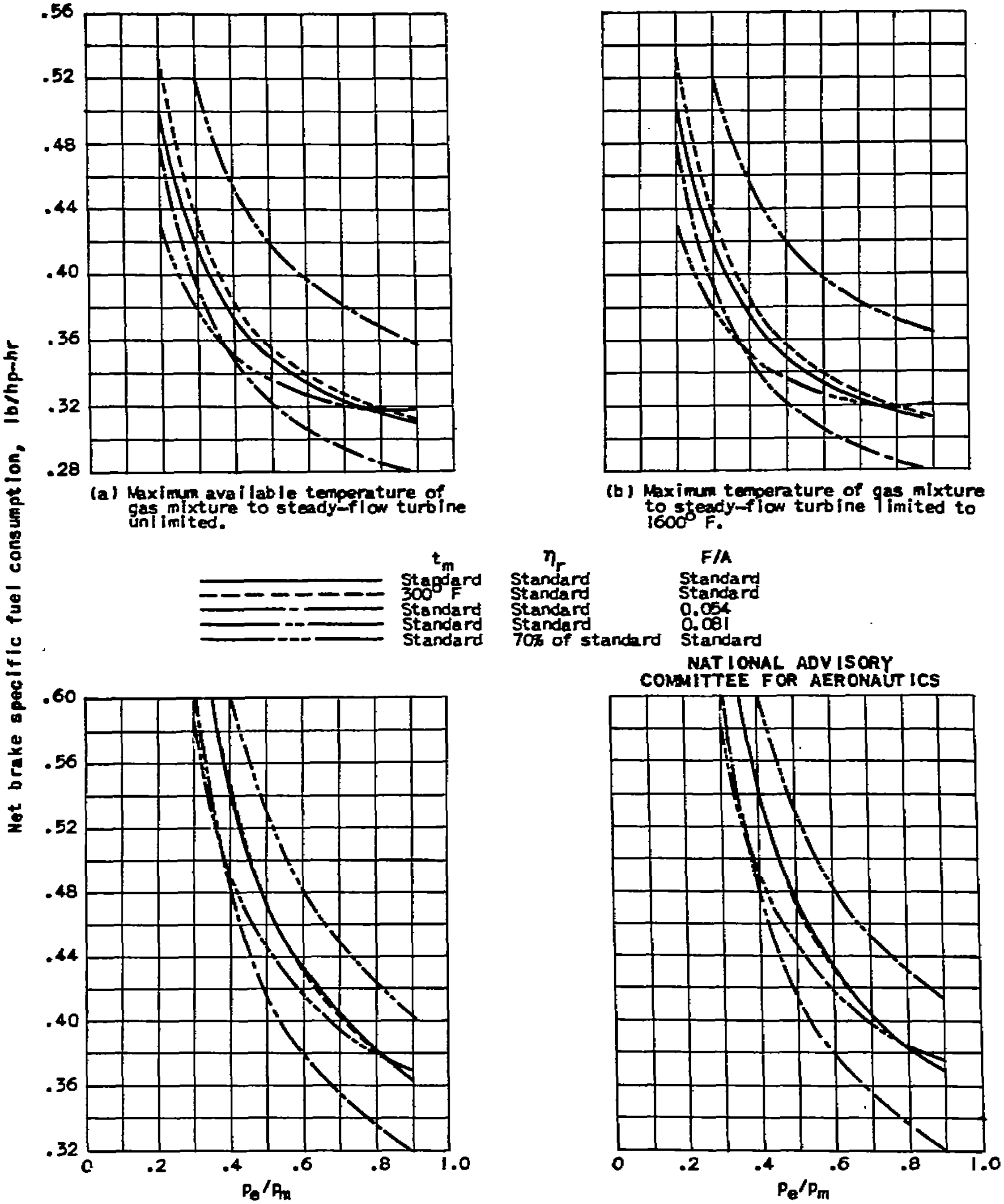


Figure 12. - Effect of change in inlet-manifold temperature, scavenging ratio, and fuel-air ratio on net brake specific horsepower of composite engine. Compression ratio, 6; standard conditions: inlet-manifold temperature t_m, 200° F; fuel-air ratio F/A, 0.067; scavenging ratio $\eta_r = 0.292\left\{\left[1-(p_e/p_m)\right]\left[t_m + 460\right]\right\}^{0.341}$



(a) Maximum available temperature of gas mixture to steady-flow turbine unlimited.

(b) Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F.

(c) Maximum available temperature of gas mixture to steady-flow turbine unlimited; blowdown turbine omitted.

(d) Maximum temperature of gas mixture to steady-flow turbine limited to 1600° F; blowdown turbine omitted.

Figure 13. – Effect of change in inlet-manifold temperature, scavenging ratio, and fuel-air ratio on net brake specific fuel consumption of composite engine. Compression ratio, 6; standard conditions: inlet-manifold temperature t_m, 200° F; fuel-air ratio F/A, 0.067; scavenging ratio $\eta_r = 0.292\{[1-(p_e/p_m)][t_m+460]\}^{0.385}$

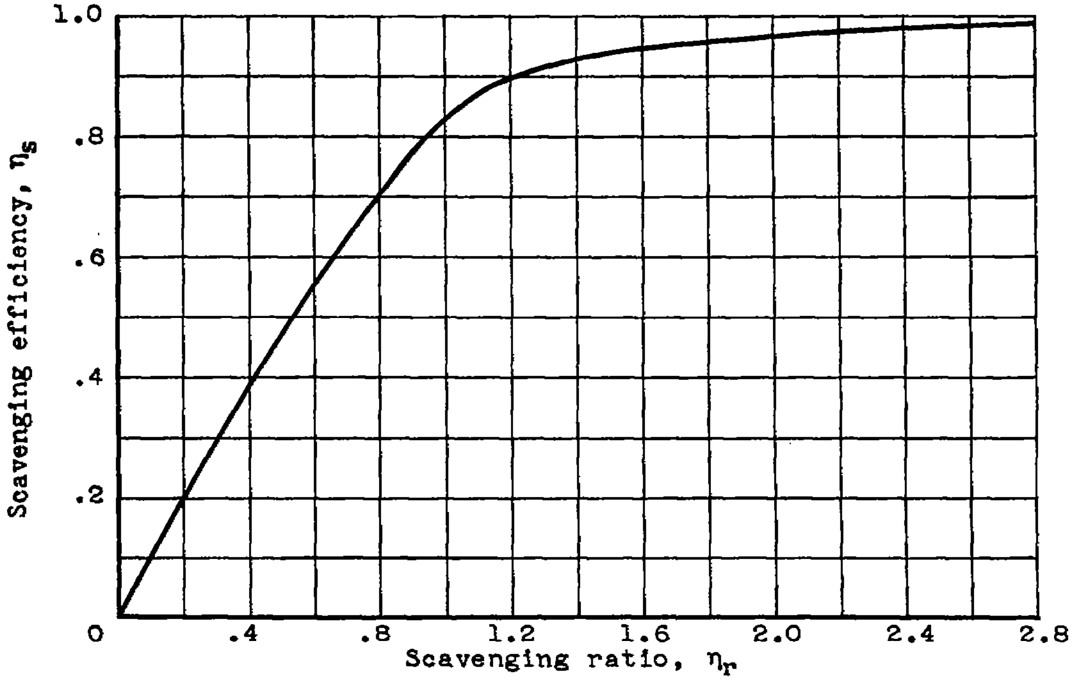


Figure 14. - Scavenging efficiency as function of scavenging ratio.

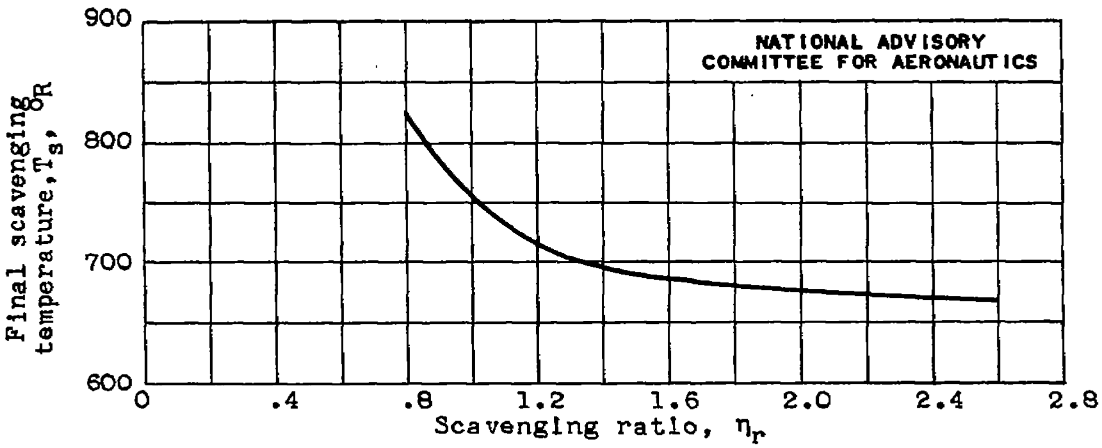


Figure 15. - Effect of scavenging ratio on temperature in cylinder at end of scavenging process.

750

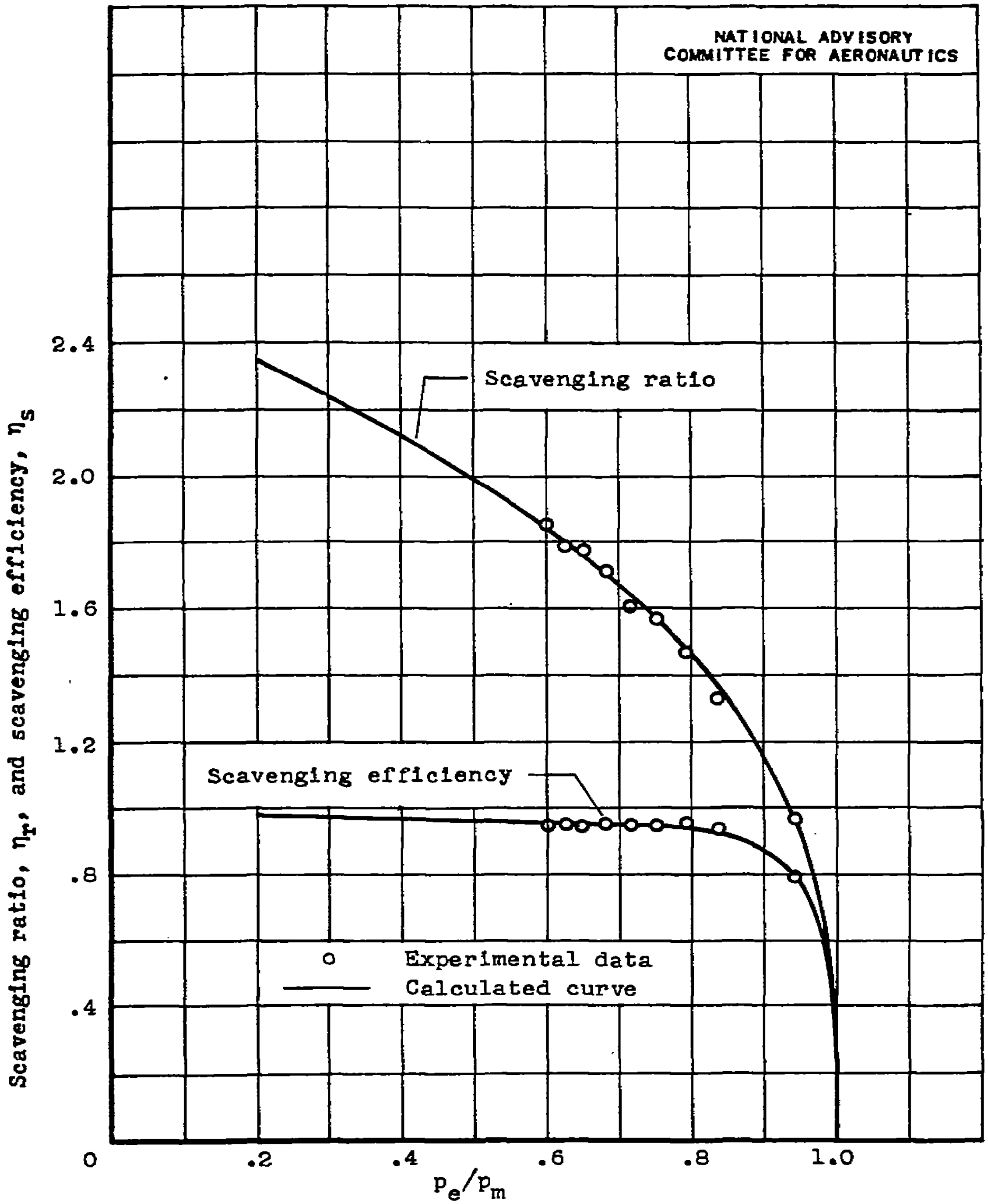


Figure 16. - Comparison of calculated scavenging ratio and scavenging efficiency with data from $4\frac{5}{8}$- by 7-inch two-stroke-cycle spark-ignition engine. Engine speed, 2000 rpm; inlet-air temperature, 100° F.

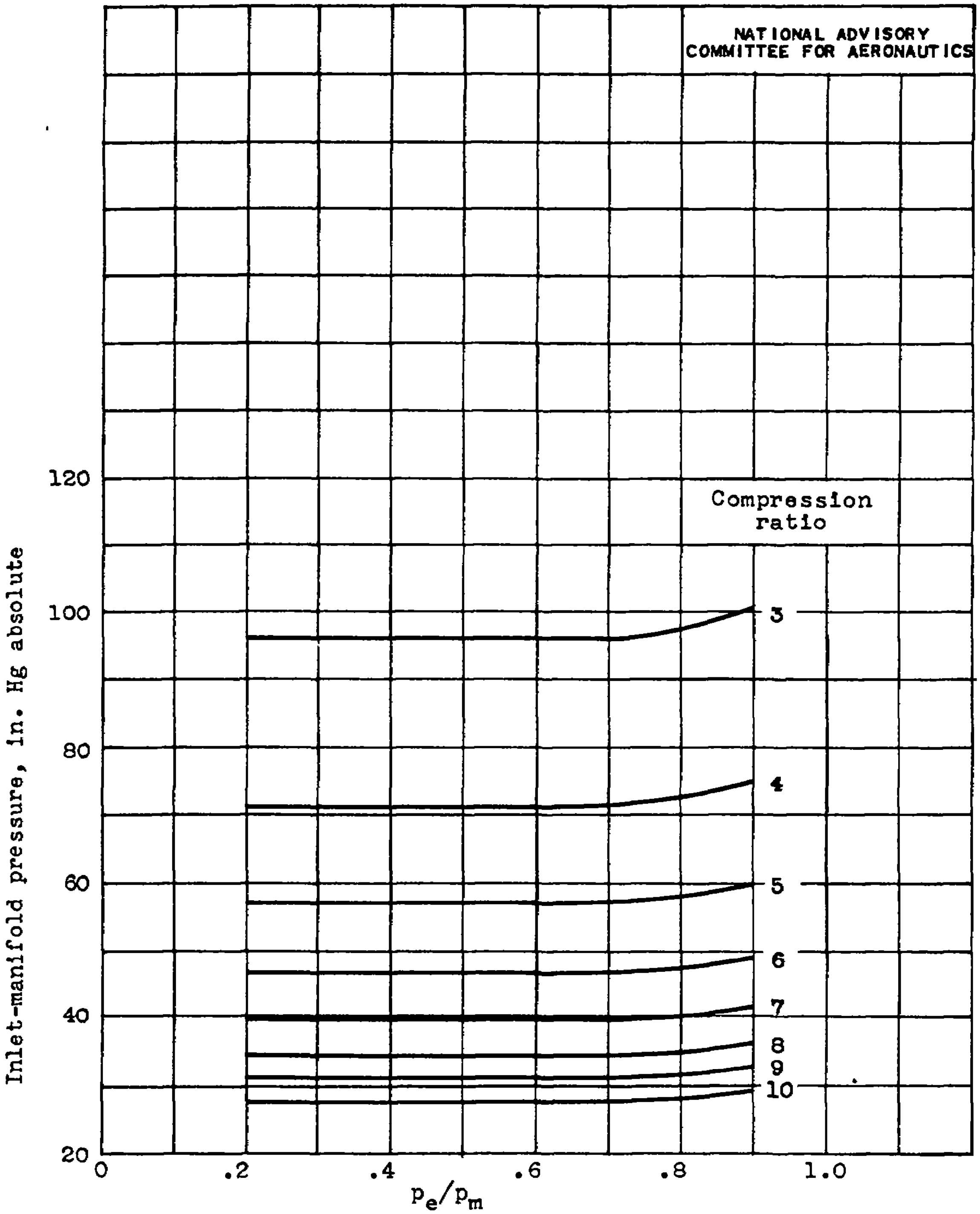


Figure 17. - Maximum allowable inlet-manifold pressure as a function of p_e/p_m and compression ratio. Maximum cylinder pressure, 1200 pounds per square inch absolute.